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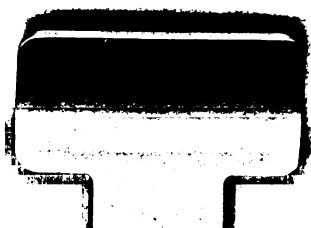
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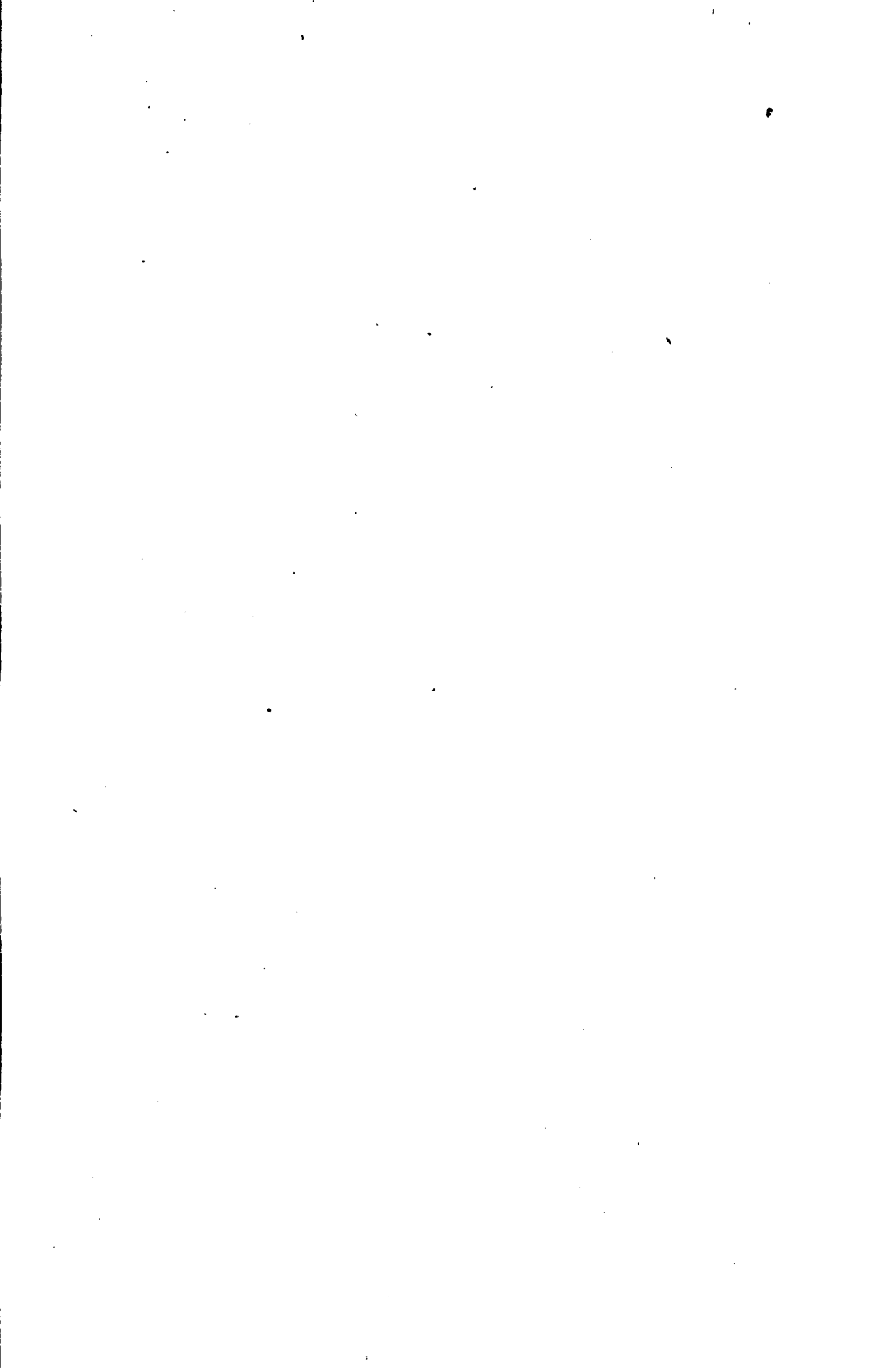
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BEARINGS AND
BEARING METALS





BEARINGS AND BEARING METALS

A TREATISE DEALING WITH VARIOUS TYPES
OF PLAIN BEARINGS, THE COMPOSITIONS AND
PROPERTIES OF BEARING METALS, METHODS
OF INSURING PROPER LUBRICATION, AND
IMPORTANT FACTORS GOVERNING THE DE-
SIGN OF PLAIN BEARINGS

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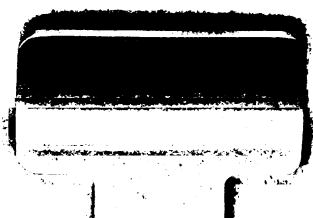
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BEARINGS AND BEARING METALS

CHAPTER I

TYPES OF PLAIN BEARINGS

BEARINGS may be divided into two general classes: journal bearings and thrust bearings. In the journal bearing the load acts at right angles to the axis; such bearings are also termed *radial* bearings. In the thrust bearing, the load acts parallel to the axis. Bearings may also be divided into two classes according to the manner in which the bearing surfaces are in contact with each other. Ordinary bearings have a sliding contact, whereas ball and roller bearings have a rolling contact. In this treatise, bearings with sliding contact only will be dealt with.

Solid and Adjustable Bearings. In the simplest form of bearing, a cylindrical shaft fits into a hole in the part which forms the support for the shaft. In a bearing of this type, there is no provision for the taking up of the wear between the shaft and the bearing. For this reason, a bushing or lining is generally used in the supporting machine part into which the shaft fits. When this bushing becomes worn, it can easily be replaced. Frequently bushings of this type are made tapered on the outside and fit into a tapered hole in the supporting frame. The bushings are split and, as the hole in the bushing wears, means are provided for pulling the bushing into the tapered hole so as to reduce the diameter of the hole. In other cases, the bushings are cylindrical, but the whole bearing is split, so that, when wear occurs, one-half of the bearing can be tightened down to restore proper conditions.

Self-aligning Bearings. When torque is applied to a shaft and the journal begins to move, the bearing should be so

supported that it will adjust itself to the journal and equalize the load over the entire bearing, thus giving an oil film of uniform thickness in a line parallel with the shaft. Small bearings may not require this self-aligning feature, the limits used being accurate enough to give uniform alignment; but larger bearings, built from a number of assembled parts, should embody this self-aligning principle. Years ago, engine builders adopted a ball-and-socket seat for this purpose. Some manufacturers prefer that design now, and, for certain classes of apparatus, there may be need for some such arrangement; but for the small limits of deflection and the higher speeds now used in machinery bearings generally, a simpler arrangement is used. An annular ring is turned in the center of the bearing shell for about one-fifth of its length, and this rests upon a corresponding support in the housing; the bearing ring extends down over the sides of the support to take up the thrust due to end play. This is a very effective and simple arrangement and is the type of support referred to when self-aligning bearings are specified in electrical machinery, for example. The other type is used for some designs, when larger limits of self-alignment are necessary, and is known as a "spherical-seat self-aligning bearing." This type is especially used for line shafting; it has a self-aligning ball-seat on the bearing, which fits into a ball-seat of the bearing box.

Characteristics of Plain Bearings. Plain bearings have been developed to meet the requirements of many different conditions of service under which shafts and spindles are required to operate.) In all plain bearings there is surface contact between the shaft and the bearing, and in order to provide for the efficient transmission of power without excessive frictional resistance, wear, and tendency of the bearing to give trouble through heating, a constant supply of clean lubricating oil must be delivered to the bearing, and this lubricant must be suitable for the conditions of bearing pressure, speed of rotation, etc. The effect of a lubricant is to prevent direct metal-to-metal contact. Some authorities explain this action by comparing the lubricant to the balls in a ball bearing. Where this comparison is made, it

is justified on the basis that the lubricant consists of a multitude of small globules of oil which roll between the shaft and its bearing, thus preventing metal-to-metal contact and greatly reducing frictional resistance.

Plain bearings are made in a great variety of designs, according to the conditions of service under which they operate, and as many of the best designs are found on machine tools, bearings for this class of service will be featured in this book. As a rule, when bearings are used to support the spindles of machine tools, where the presence of any considerable amount of lost motion would result in inaccuracy of work produced by the machines, means of adjustment are provided to take up any lost motion that develops as a result of wear in the bearing. In the detailed descriptions of different types of plain bearings, the manner in which designers have worked out means of compensating for wear will be explained.

Adjustable Bearings for Machine Tool Spindles. In designing the bearings for carrying the spindles of machine tools, it is very important to provide means of constantly maintaining a tight fit between the spindle and its bearing boxes, in order effectually to eliminate chatter and vibration. It is evident that any lost motion in the bearings will seriously affect the accuracy of work produced on the machine and the perfection of finish which it is possible to obtain on the work. Not only must the bearings be a perfect fit at the time the machine is new, but provision must also be made to compensate readily for any wear which develops after the machine is placed in service, so that it will continue to produce work of the required degree of accuracy. To meet the requirements of this service, some form of tapered bronze bearing has been quite generally adopted as a standard spindle bearing construction, although various modifications of this general form of design have been worked out by different machine tool builders to provide for making the necessary adjustment for wear. These modifications may be made to meet the requirements of different conditions of service for spindles which are mounted in a vertical or a horizontal position, or they may be worked out

simply to meet the individual opinions of different designers. In the following discussion, there are presented descriptions and illustrations of different well-known forms of adjustable spindle bearing designs, which have been found to give satisfactory service.

Grinding Wheel Spindle Bearings. In order to give satisfactory service, the spindle of a grinding machine must rotate without an appreciable amount of vibration; otherwise

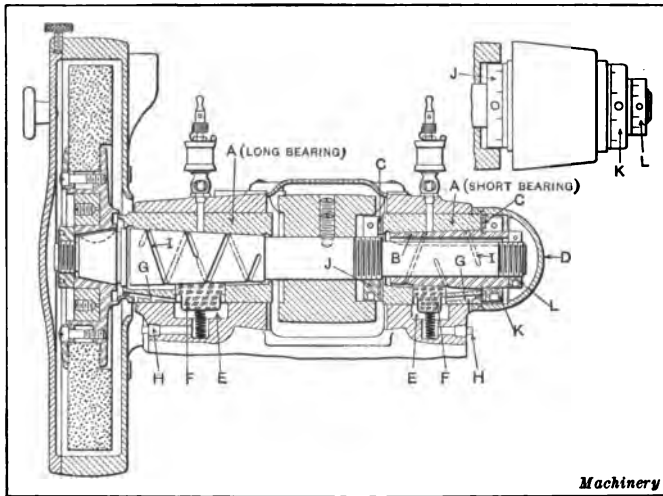


Fig. 1. Design of Spindle Bearings for External Grinding Machine, showing Provision made to compensate for Wear and Method of Lubrication

chatter marks will appear on the work. It is safe to say, perhaps, that in working out the design of the average grinding machine, more time is given by the designer in developing a satisfactory form of spindle bearing construction and in modifying this design to overcome trouble which is experienced when the first machine is placed in service, than in producing satisfactory results in any other part of the mechanism. In Fig. 1 there is shown the standard type of grinding wheel spindle bearing which has been adopted for use on external grinding machines built by the Landis Tool Co., of Waynesboro, Pa. The bronze bushings *A* are assembled in the housings in such a way

that expansion and contraction, due to changes in temperature, do not in any way affect the transmission efficiency of the bearings; and compensation for wear provided by the tapered form of the bushings enables all lost motion to be taken up, thus effectually eliminating vibration and enabling the grinding machine to produce perfectly accurate work. Attention is called to the fact that adjustment for wear in the front and rear spindle bearings is made independently, so that if only one bearing requires adjustment, this can be done without touching the other bearing. On these bearings, lubrication is provided by individual sight-feed oil-cups, from which a supply of oil is delivered to a reservoir *E* beneath each of the bearings. Felt wicks take oil from these reservoirs and carry it up to the bearings, the wicks *F* being held in contact with the bearings by means of compression springs. It will be seen that each reservoir is provided with a threaded plug *H*, which may be removed to drain out dirty oil and flush the bearing with kerosene before the plugs are replaced and a fresh supply of lubricating oil is delivered to the reservoir. Drains *G* prevent the possibility of reservoirs *E* overflowing and flooding the bearings. Oil-distributing grooves *I* are cut in the journals to facilitate the uniform distribution of oil. The way in which these spindle bearings are adjusted will be apparent from the illustration. Bronze bushings *A* are a press fit in the housings, and journal sleeve *B* is fitted over the back end of the spindle. The thrust load is carried by steel thrust washers *C*. When it is necessary to compensate for wear in the journal bearings, this is done by releasing collar *J* and tightening *L* to draw the front bearing into its box to take up lost motion; similarly, lost motion in the rear bearing is taken up by releasing collar *K* and tightening *L*, which forces sleeve *B* into the tapered bronze box. Each division on collars *J*, *K*, and *L* gives 0.00052 inch of adjustment on the diameter of the bearings.

Self-aligning Spindle Bearings. Features of particular interest in connection with the design of bearings for the wheel-spindle of a Brown & Sharpe grinding machine,

shown in Fig. 2, are the provisions made for lubrication and for the automatic alignment of the bearings with each other. It will be seen that a sight-feed oil-cup is provided at the top of each bearing housing, from which lubricant is delivered to the annular channel *A* cut in the inside of sleeves *D*. Oil runs around this channel and is absorbed by a felt packing or wick *B*, which is contained in a slot cut through the bronze bearing box on the under side of the shaft. This felt packing is kept constantly saturated with

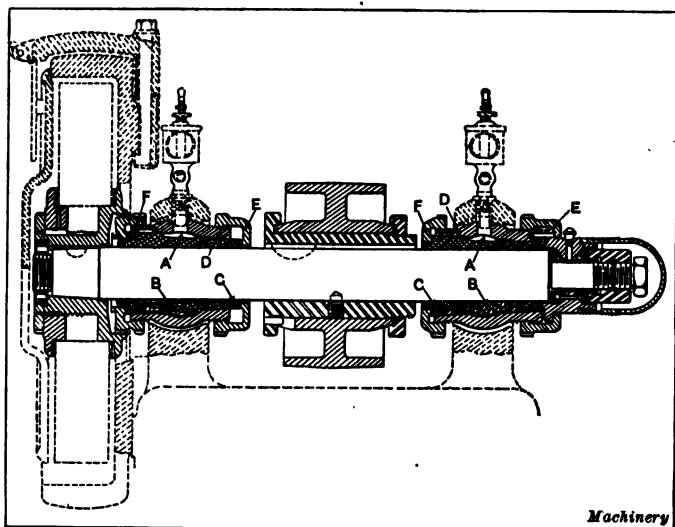


Fig. 2. Another Example of Bearing Construction for Grinding Machine Spindle

oil, and is held against the shaft by a flat spring. As the shaft revolves in the bearing box, the felt distributes a uniform film of oil over the shaft, thus efficiently lubricating the bearing.

To assure accurate alignment between the two bearings, each of the bronze boxes *C* is carried in a sleeve *D* that is machined to a spherical form on the outside in order to fit into a corresponding spherical seat in the bearing housing. When a straight shaft is put through bronze bearing boxes *C*, it will be apparent that the bearings will align themselves accurately on account of the ability of the spherical-seated

sleeves *D* to move as required in the bearing housings. The bronze bushings *C* in these bearings are tapered on the outside and bored straight on the inside; this is not quite so

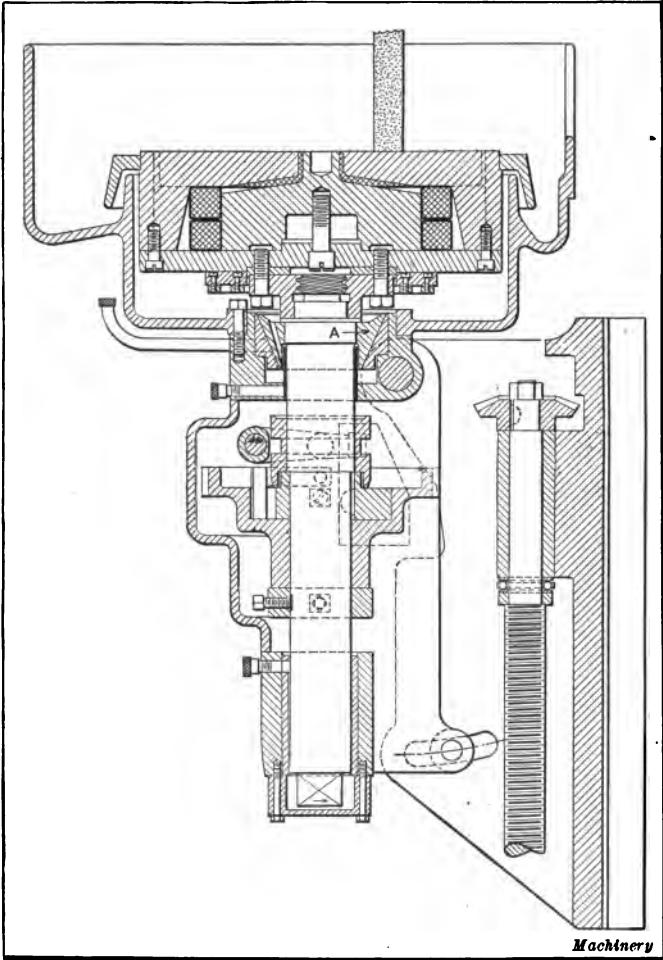


Fig. 3. Bearing for Table of Rotary Surface Grinding Machine, showing Method of automatically compensating for Wear

usual an arrangement as that of bronze boxes which are bored to fit a tapered journal on the shaft they are designed to carry. These boxes *C* are split, and in order to compen-

that the bearing runs quite freely and with very little friction. There is said to be absolutely no chance for side play in this spindle bearing which would cause the work-table or magnetic chuck to get out of alignment in any direction. Collar *A* and the bronze box in which it runs are arranged to form an oil reservoir from which a constant supply of lubricant is delivered to the bearing, lubrication being facilitated by spiral grooves in the taper collar which carry the lubricant and distribute it uniformly over the rubbing surfaces.

Adjustable Bearing Box of Straight Cylindrical Form. Still another form of adjustable spindle bearing box is shown in Fig. 4. This is another Brown & Sharpe construction, which has been designed for use in headstock spindle bearings. It will be seen that this box differs from any of the designs which will be shown in this chapter in that it is of straight cylindrical form on both the inside and outside. The method of making adjustment of the fit of this box on the spindle will be best understood by referring to the detailed view of the box shown at *A*, where it will be seen that a split extends all of the way along one side. Two dovetail slots *B* are machined in this split section to accommodate screws and beveled nuts or sleeves which may be tightened or loosened in order to expand the opening in the box or cause it to be contracted through pressure applied by the cap of the bearing housing. Reference to the end view will show that there are two screws *C* which hold the cap of the bearing housing down on top of the bronze box, and when it is desired to secure a tighter fit of the box on the spindle, this result is accomplished by first backing away the screws in the sleeves in slots *B* and then tightening up the screws *C* on the cap of the bearing housing. In this way exactly the required fit for the bearing box may be obtained, and after the bearing has been in service, compensation for wear may be accomplished by further adjustment of screws *C* and the screws which adjust the sleeves fitting into slots *B*.

Examples of Milling Machine Spindle Bearing Design. For supporting the spindle of the vertical milling machine

built by the Garvin Machine Co., Spring and Varick Sts., New York City, the following construction has been adopted: At the lower end of the spindle, Fig. 5, there is a bearing box *A* which is bored to a taper corresponding to the tapered journal on the spindle, to provide means of taking

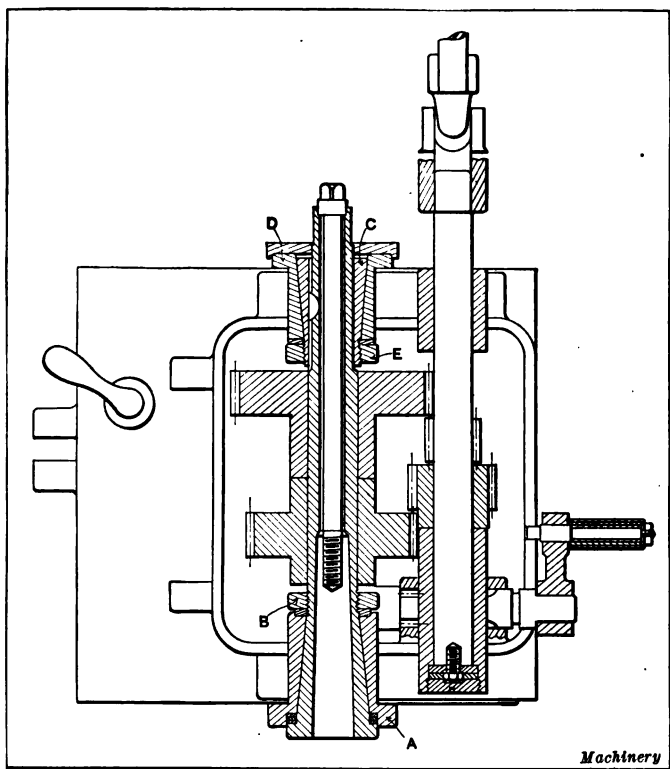


Fig. 5. Design of Adjustable Bearings for Vertical Milling Machine, showing Means of Compensating for Wear

up any wear that may develop. It will be seen that the spindle is threaded along the straight portion immediately above the tapered journal to receive a threaded collar *B* which provides for holding the spindle up. When lost motion develops it is easily taken up by simply pulling the spindle up into the tapered bearing by tightening collar *B*. To provide for making this adjustment, it is necessary to

face off the soft washer between bearing box *A* and the spindle flange. At the upper end, the spindle is straight and is carried in a bushing *C* which is bored straight on the inside and turned to a taper on the outside, this bushing being secured to the spindle with a Woodruff key so that it is a tight fit. On the outside, bushing *C* is turned to a suitable taper to fit bearing box *D*, and bushing *C* is threaded at the lower end and furnished with a collar *E* which provides for maintaining a tight fit between bushing *C* and bearing box *D*. By screwing up collar *E*, provision is made for taking

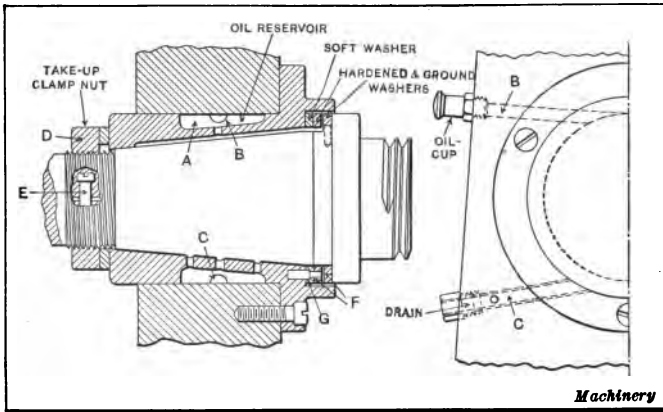


Fig. 6. Adjustable Bronze Spindle Bearing furnished with Means of compensating for Wear and Provision for Continuous Lubrication from a Reservoir

up any wear which develops in this bearing after the machine is put into service. It will be apparent that the use of bushing *C*, which is keyed to the spindle, is a substitute for having the journal made an integral part of the spindle, as it is at the lower end. The reason for having the upper journal in the form of a bushing keyed to the spindle is that this greatly facilitates the work of assembling the machine. Collars *B* and *E* are provided with split nuts to hold them in place after making the settings.

Provision for Cleaning Bearings. Fig. 6 shows a form of spindle bearing construction which has been adopted as one of the standards in building machine tools made by the Garvin Machine Co. It will be apparent from the illustration

that in working out this design provision has been made for supplying the bearing with lubricant from reservoir *A*, which extends around the outside of the bearing box. A supply of oil is delivered into this reservoir through tube *B* and finds its way from the reservoir to the bearing surface through holes which connect with oil-grooves cut on the inside of the bearing box. The reservoir is cleaned out by opening drain pipe *C* to allow the dirty oil to run out, after which the bearing is flushed with kerosene before closing drain *C* and filling the reservoir with a supply of fresh oil. It will be seen that the spindle is threaded on the straight portion directly to the left of the tapered journal, to provide for drawing the spindle into its tapered bearing box when it becomes necessary to compensate for wear. Before this adjustment can be made, soft washer *G* must be faced off. Collar *D*, by means of which this adjustment is made, is split at one side and furnished with a clamp screw *E*, so that when the bearing has been adjusted to obtain the

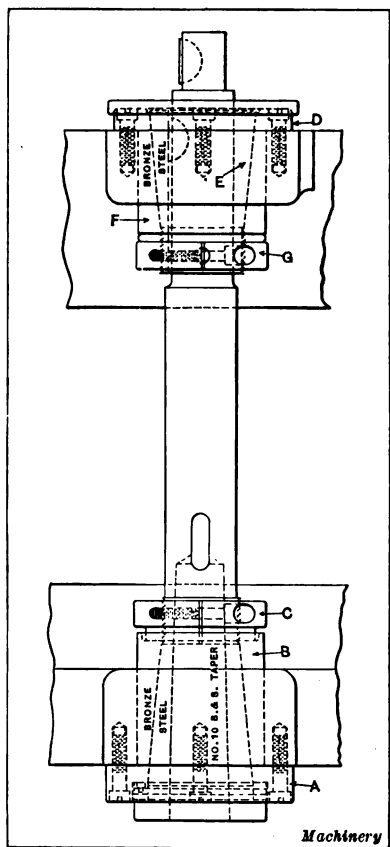


Fig. 7. Adjustable Bearings designed for Vertical Spindle Profiling Machine

desired fit, screw *E* is tightened to clamp collar *D* firmly in the exact position to which it has been set. In most machine tool bearings, provision must be made for carrying a thrust load in addition to the normal radial load which comes on the bearing. In the present case, this result is

obtained by designing the spindle with a flange at the large end of the taper. This flange bears against two hardened and ground washers *F* and a soft washer *G*, which are placed between the flange on the spindle and the end of the tapered bearing box, the thrust load being carried in this way without imposing an undue strain and unnecessary amount of wear on the bearing.

In Fig. 7 there is shown the form of adjustable bronze bearings which are used for carrying the vertical spindle of the Nos. 2 and 4 profiling machines built by the Garvin Machine Co. As in the case of the bearing design shown in Fig. 6, it will be seen that provision is made for carrying

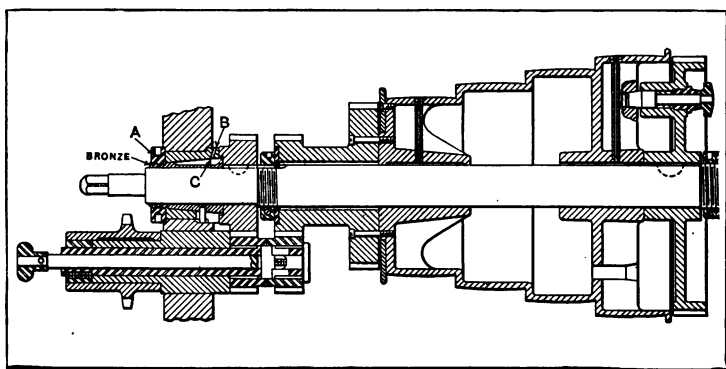


Fig. 8. Example of Bearing Design where Compensation for Wear is accomplished by compressing the Bronze Bearing Box

the thrust load by having a flange at the lower end of the spindle which bears against an arrangement of thrust washers. These washers serve the double purpose of taking the thrust of the spindle and providing means of compensating for any wear and lost motion which may develop in the lower spindle bearing.

When wear in the bearing must be compensated for, this is done by facing off the soft thrust washer and tightening collar *C*, which is carried on the threaded section of the spindle, the result being that the spindle is drawn up into the tapered bearing box. At the upper end of the spindle there is a tapered steel bushing *E* which is keyed to the straight end of the spindle with a Woodruff key. On the

outside, bushing *E* is machined to a taper to fit bronze bearing box *F* accurately, and when it is necessary to make adjustment to compensate for wear, threaded collar *G* is tightened. It will be apparent that both collars *C* and *G* are split on one side and furnished with binding screws to provide for securing the collars in the desired positions after the required adjustment has been made.

Compensation for Wear by Compressing Bearing Box.

In Fig. 8 there is shown a different method of providing compensation for wear in a tapered bronze spindle bearing box.

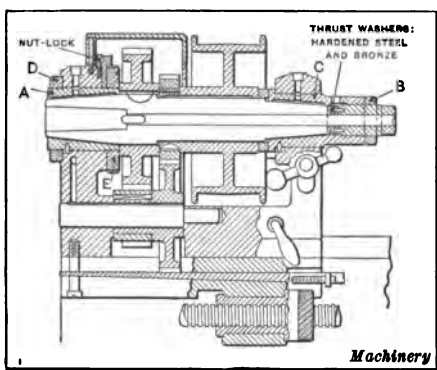


Fig. 9. Bearings designed so that Compensation for Wear is accomplished by drawing Spindle Into Tapered Box

This construction is used on the Garvin No. 2 universal milling machines. At the front end, the spindle is furnished with a bearing box of the same general design as those which have already been described, but at the rear end, as will be seen, the bronze box is split and fitted to the straight end of the

spindle, this box being tapered on the outside to engage a bushing in the machine frame with which it remains in fixed contact. This necessitates an entirely different method of adjusting the box to afford compensation for wear, and such adjustment is obtained by tightening threaded collar *A* to provide for pulling the tapered bearing box into the bushing in the machine frame. When this collar is tightened to pull the bearing box into the tapered bushing, the bronze box is compressed against the straight journal on the spindle, thus taking up any wear which may have developed. Attention is also called to oil-hole *B*, which delivers a supply of oil into reservoir *C*, formed in this bearing box to provide a liberal supply of

oil; the oil flows from the reservoir through oil-holes connecting with grooves machined in the bearing surface of the box to provide for distribution of lubricant over the bearing. The same arrangement of an oil-hole and oil reservoir in the bearing box is provided at the front end of the spindle.

Compensation for Wear by Drawing Spindle into Tapered Box. Fig 9 shows an example of the design of bearings for a machine tool spindle, where the journals are directly supported by tapered bronze boxes at both ends. It will be seen that at the front end of the spindle (which is the left-hand end in this case) a tapered bronze box *A* is provided which is threaded on the outside at both ends. To compensate for wear which may develop in these bearing boxes, the method of procedure is as follows: Threaded collar *B*, carried at the right-hand end of the spindle, is tightened to draw the entire spindle, and hence the tapered bearing at this end of the spindle, back into box *C* to compensate for any lost motion which may have developed as the result of wear. When this result has been obtained, a corresponding adjustment must be made on box *A*, the procedure being as follows: Jam nut *B* is loosened and the rear step plug behind the nut is slacked away, after which threaded collar *D* at the front end of the spindle is loosened and collar *E* is then tightened to draw the front tapered bearing box up to a snug fit on the journal; this also pushes the entire spindle back into the rear tapered bearing *C*. When this result has been obtained, collar *D* is tightened to hold the box in exactly the position in which it has been set. Then the rear step plug is carefully adjusted against the rear end of the spindle and locked by a jam nut *B*. This is another bearing design used by the Garvin Machine Co.

Cast-iron and Babbitt Spindle Bearings. Experience has shown that very satisfactory service is obtained from a hard steel journal running in a cast-iron bearing. With such a combination, it is found that the cast iron has a tendency to become glazed on the surface in such a way that the coefficient of friction between the bearing and its journal is

very small, thus making the transmission efficiency of the bearing correspondingly high; also a bearing of this kind is so hard that wear becomes almost a negligible quantity. In Fig. 10 there is shown a type of spindle bearing construction which has been adopted as a standard by the R. K. LeBlond Machine Tool Co., of Cincinnati, Ohio, in building lathes, milling machines, and cutter grinders. This consists of a cast-iron box at the front end and a babbitted

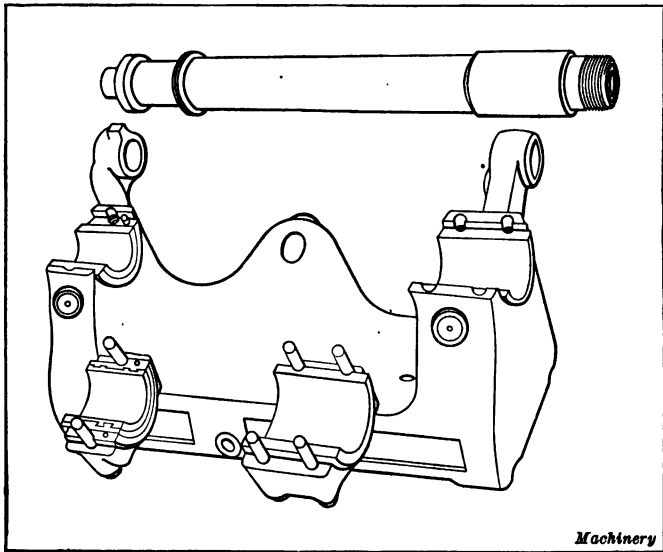


Fig. 10. Example of Spindle Construction used for Lathes, Milling Machines, and Cutter Grinders, where Front Bearing is of Chilled Cast Iron and Rear Bearing is lined with Babbitt

box at the rear end of the spindle, and the results obtained by this combination have proved so satisfactory in practice that a detailed description will be of interest.

The spindle is made of 50-point carbon steel, and over this there is pressed at the front end a hardened steel bushing made of Shelby seamless tubing. In making this bushing, the blank is cut off and annealed, after which it is bored, turned, and carburized; then it is necessary to round up the bushing to remove any distortion which has been

produced during the heat-treatment, after which it is reheated and quenched in water to make it glass-hard. After hardening, the bushing must show a scleroscope hardness of 80 degrees, and those which fail to fulfill this test are heat-treated again. The hardened bushings are ground on the inside and rough-ground on the outside, after which they are again tested for hardness before being pressed on the spindle. The fit of the bushing on the spindle is so adjusted that from seven to ten tons' hydraulic pressure is required to push it into place, after which the bushing is ground to standard size. The cast iron for the bearing is cast against a chill to make the metal dense and close-grained. Mention has already been made of the fact that this hardened steel journal is carried in a cast-iron bearing box which is an integral part of the main headstock casting. This eliminates an extra joint between the bronze box or a babbitt-lined bearing, and affords a more rigid construction, although the work of scraping the cast-iron spindle bearing to an accurate fit and perfect alignment is a difficult job, calling for the services of an experienced mechanic. A cast-iron bearing of this type must be more accurately finished than either a babbitted or a bronze-bushed bearing, because no dependence can be placed upon the spindle wearing itself to a satisfactory fit after the machine is placed in service; the bearing must fit properly before the machine is started. Several years ago the R. K. LeBlond Machine Tool Co. built four trial head-stocks equipped with the following combinations of journals and bearings: (1) Cast-iron bearings and soft steel spindle; (2) bronze bearings and soft steel spindle; (3) babbitted bearings and soft steel spindle; and (4) cast-iron bearings and hardened steel spindle.

Lathes with spindles of these types were kept in constant use on work of the same general character, and when examined, the condition of the hardened steel journal running in a cast-iron box was found to be the best, neither the spindle nor the box being worn to an appreciable extent and the grinder and scraper marks still being visible. The

combination of a soft steel journal and bronze box was in good condition, but the journal was slightly ridged in the center. The soft steel spindle in a cast-iron box was appreciably worn, but the soft steel spindle in a babbitt box was in first-class condition. There are lathes in the Le Blond shops at the present time in which hardened steel spindles have been running in cast-iron boxes for about twelve years without any adjustment having been necessary. The wear is so slight that it can scarcely be measured with the most delicate instruments. It is very important, however, to have the combination of a hardened steel spindle and cast-iron box, as a soft steel spindle and a cast-iron box is much less satisfactory than a soft steel spindle and either a bronze-bushed or babbitted bearing.

At the rear end, the spindle of LeBlond lathes, milling machines, or cutter grinders is supported in a babbitted bearing. It will be seen that the seat for this babbitted bearing is dovetailed and the babbitt is poured, after which the bearing is bored in position. No attempt is made to peen the metal. Experience has shown that wear on the back bearing in a headstock of this construction is proportionate to the wear of the front box with its hardened steel journal running in a cast-iron bearing. Practically no attention is required by bearings of this form after the front bearing has run for a sufficient time to enable the cast iron to become glazed. The metal becomes so hard on the surface that it can scarcely be "touched" with a scraper.

Oilless Bearings. The advantages of the oilless or self-lubricating bearing for many classes of service have led to the development of several different types, each of which doubtless has its advantages when applied under suitable conditions. One type consists of wood impregnated with wax, oil or paraffin. Another is made of bronze and has graphite inserts. Still another type is formed of graphite impregnated with some bearing metal such as a white metal alloy or bronze.

One type of "oilless" bearing has on the bearing surface, symmetrical grooves of various designs, depending upon

the service for which the bushing is intended. These grooves are packed solid by means of hydraulic pressure with a special hard graphite lubricant. When the bushings are once installed and in use, fine particles of the graphite lubricant are distributed over the entire bearing surface of the bushing. Ample lubrication is provided for almost any form of bearing contact, whether the movement be oscillating, vertical, horizontal, or full revolution. Owing to the scientific design of the graphite grooves which fits them for the particular service required, wear is reduced to a minimum. The design of the graphite-packed grooves varies in different bushings because some bushings are subjected to different kinds of contact from others. These bushings are suitable for use in machine tools, countershafts, lineshaft hangers, electric motors, etc., but, although they are styled "oilless" bearings, it is recommended that about 25 per cent of the volume of oil which would be required to operate efficiently an ordinary plain bearing of the same size should be applied to these oilless bearings. However, in the event of failure on the part of the attendant to give the bearing the required oil, this lack of attention is not so likely to result disastrously, as would be the case with an ordinary bronze-bushed bearing.

Another type of oilless bearing consists of a bushing made of either hard maple or iron wood, which is thoroughly impregnated with a mixture of graphite and other lubricant by a special treatment. These are more truly "oilless" bearings than the type that has graphite inserts, because when the bearing is in operation a slight increase in temperature resulting from friction causes the bearing to exude some of the lubricant with which it has been impregnated, thus providing for efficient transmission of power. These bearings are especially adapted for use in loose pulleys; and in many cases they are run without any provision for lubrication. Some users prefer to drill the usual oil-hole in the hub of the pulley and in the bushing, so that a little oil can be added from a squirt can, and very satisfactory results were obtained where the bushings were

used in this way. They must be properly supported to prevent splitting, although with sufficient support these wooden bushings have ample strength to resist any crushing load to which they will be subjected when used in loose pulleys and similar positions.

Still another type of oilless bearing consists of graphite bushings made of the required sizes, and impregnated with white metal or bronze to produce what is known as "graph-alloy." The metal used for impregnating the graphite depends upon the class of work. All parts such as shaft bearings are impregnated with a white-metal alloy or babbitt metal of special composition. The parts used in connection with electrical apparatus are impregnated with a copper alloy which is also utilized for such parts as steam turbine packing rings, etc., which must withstand high steam temperatures. The use of copper for the electrical work is essential because of its electrical conductivity.

The extent to which the graphite is impregnated with the metal is indicated by the fact that the weight of the graphite is 0.057 pound per cubic inch, whereas the weight of the metalized product is 0.145 pound per cubic inch when impregnated with babbitt, the increase of weight due to the metallizing process being practically 150 per cent. The metal in "graphalloy" is about 60 per cent by weight, or 25 per cent by volume. The specific gravity is 4 and the compressive strength, approximately 14,000 pounds per square inch. "Graphalloy" is not injured by the application of lubricant. In fact, the use of an applied lubricant is recommended when the bearings are used for rather heavy service.

At the present time "graphalloy" in the form of bearings is applied to light-duty machinery operating at high speeds and to heavier service when the speeds are relatively low. Its use is recommended particularly where the application of oil is either objectionable, difficult, or likely to be neglected. These bearings are commonly applied to loose pulleys, vertical shaft bearings, conveyors, textile machin-

ery, canning machinery, etc. They are recommended for loose pulleys in cotton and silk mills and have also been utilized successfully on paper machinery, shoe machinery, wrapping machines, cotton spinning frames, etc., which operate on a product that may be damaged by the use of oil. One of the important uses is for vertical bearings, such as are found on governors, fans, etc. The use of "graphalloy" is not recommended where the combination of speed and pressure is excessive. As a general rule, a bearing pressure of 50 pounds per square inch of projected area should not be exceeded. The limitations of this material are indicated by the following rule which applies to a bearing used without a lubricating oil: The surface speed of the shaft in feet per second multiplied by the pressure in pounds per square inch on the projected bearing area should not exceed a constant of about 200. The temperature of "graphalloy" bearings is somewhat higher than that of ordinary bearings lubricated with oil which is principally due to the fact that "graphalloy" is a relatively poor conductor of heat. As soon as a "graphalloy" bearing is put into service, a graphite coating begins to form on the shaft, the thickness of the coating depending upon the smoothness of the shaft. This coating soon becomes very smooth and hard, which greatly reduces the coefficient of friction and the temperature of the bearing. It is claimed that a temperature as high as 200 or 300 degrees F. will not injure the bearing or cause it to seize the shaft. The temperatures are moderate at high speeds, provided the bearing pressure is low. On the other hand, if the bearing pressure is relatively high and the speed low or moderate, the temperature will also be low.

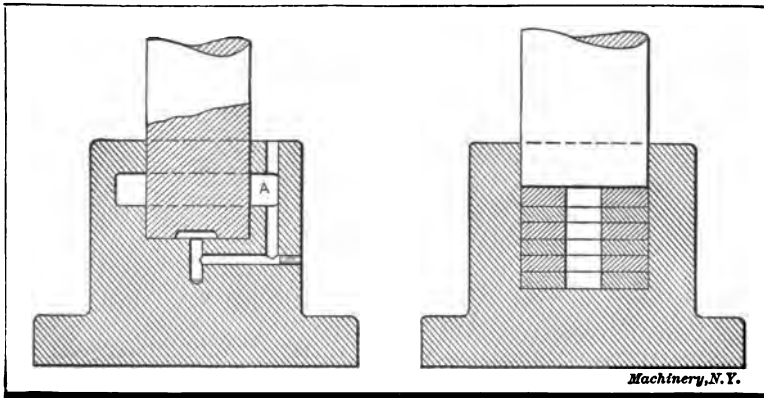
Thrust Bearings. Thrust bearings are of two very general classes: step bearings and collar bearings. In step bearings, the thrust is taken by the end of the supporting shaft; in collar bearings, by projections or shoulders. The simplest kind of a thrust bearing is the pivot bearing, exemplified by the bearings for watch pinions and by a lathe

center taking the end-thrust of a cut on a piece held between the centers. In general, however, the end-thrust is taken by a large flat or nearly flat surface. When this is the case, several considerations present themselves which must be given due attention by the machine designer. Assume that the flat end of a vertical cylindrical shaft carrying a weight or otherwise subjected to pressure is supported by a flat surface. Then, if the shaft rotates, the velocities of points on its end surface at different radial distances from its axis, will vary. The velocities of the points near the outside will be, in comparison, very high, while the velocity of a point near the center will be low. On account of this variation in velocity, the wear on the end surface of the shaft and the thrust surface of the bearing will be considerably uneven. If the parts are well fitted together when new, so that a uniform pressure is produced all over the end of the shaft and bearing, then the outer parts of the bearing surfaces will wear away most rapidly. This again increases the pressure at the center, which sometimes may become so intense as to exceed the ultimate crushing strength of the material. The unequal wear of the surfaces of thrust bearings is one of the most difficult problems meeting the designer of machinery of which such bearings form a part.

The Schiele Curve. Experiments carried out by Schiele show that the wear is theoretically along a curve called the *tractrix*. If an end-thrust bearing is made of a form corresponding to the Schiele curve then the wear in the direction of the axis of the thrust shaft will be uniform at all points; but while this curved form would be theoretically correct it has been shown in practice that nothing is to be gained by the use of bearings having this complicated shape.

Simple Step Bearings for Light Duty. For light duty, simple step bearings of the types shown in Figs. 11 to 14 meet the requirements. The intense pressure at the center and the consequent unequal wear are partly avoided in the

bearing in Fig. 11 by cutting away the metal at the center of the shaft, as shown, leaving an annular ring which takes the thrust. This procedure is advisable in all step bearings. Another difficulty met with in bearings of this type is the question of lubrication. If the speed of the shaft is high, the centrifugal force tends to throw the oil out from the center. Special provisions must then be made for again returning the oil to the center, as otherwise the bearing would wear down rapidly, become heated, etc. In Fig. 11, a simple method is shown for automatically returning the oil to the bearing surfaces. An oil-passage is made from



Figs. 11 and 12. Simple Designs of Step Bearings

the chamber A formed around the shaft to the center of the shaft at the bottom. When the channel and chamber are once filled with oil, this oil will continue to circulate automatically; it will be drawn in at the bottom, be thrown outward by the centrifugal force, find its way into the chamber A, and finally, through the channel, return to the center of the bearing.

When a bearing for heavier duty is required, the design shown in Fig. 12 is quite commonly adopted. A number of disks or washers are placed between the end of the thrust shaft and the supporting bearing, in order to introduce a number of wearing surfaces, instead of having the

end of the shaft and the box take all the wear. Due to the fact that the series of washers introduced permits of a lower speed between each pair of washers, the wear is quite materially reduced. Should the pressure cause any two washers to heat and bind, the frictional resistance between them ceases, as one washer is free to follow the motion of the other, and the oil will have an opportunity to get between the surfaces and cool them off.

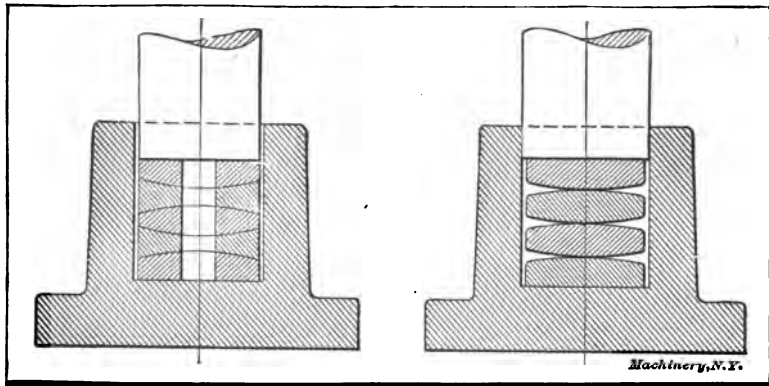
A hole may be, and generally is, drilled through the centers of the washers, as shown in Fig. 12, and the same method for continual lubrication, as shown in Fig. 11, may be used to advantage. Every alternate washer is commonly made of hardened tool steel or case-hardened machine steel, while the others are made of bronze. This combination provides for good wearing qualities. If the thrust shaft is made of soft machine steel, and the box of cast iron, the top washer is often secured to the shaft, and the bottom washer to the box, so that all the wear may be concentrated upon the washers, which can easily be replaced.

In Fig. 13 is shown an improvement on the bearing in Fig. 12. This construction is recommended, in particular, in cases where the shaft and its bearing box cannot be properly aligned with one another. The washers have spherical faces, being alternately convex and concave. They are slightly smaller in diameter than the bearing box into which they are inserted, so that they may have an opportunity to adjust themselves to a perfect bearing on each other, and thereby make up for the differences in the alignment of the thrust shaft and bearing box.

Another type of thrust bearing for loads which are not excessive is shown in Fig. 14. It is a well established principle that it is better to take the thrust of a bearing as near the center of the shaft as the load to be carried will allow. The farther away from the center the support is, the greater is the motion and the greater is the retarding effect of the friction. The thin convex washers used are of tool steel,

hardened, and, although the bearing between them is very small, their strength and hardness is such that they are capable of standing a considerable pressure, although not so great a one, probably, as the other forms shown in Figs. 11, 12, and 13. In this bearing, also, there is no difficulty in keeping the surfaces well oiled, since all that is necessary is to keep the chamber well flooded with oil.

As already mentioned, flat thrust bearings should be made of an annular form. It is good practice to make the inside diameter one-half of the external diameter. Experiments made on flat pivot thrust bearings, three inches in



Figs. 13 and 14. Other Designs of Step Bearings

diameter, indicate that the coefficient of friction between a steel pivot and a manganese-bronze bearing, properly lubricated, using two radial oil grooves only, varies from 0.018 at 50 revolutions per minute, to an average of 0.011 at 350 revolutions per minute. If four radial oil grooves are used instead of two, the friction is approximately doubled, due to rupture of the oil film.

Load on Thrust Bearings. The load that may be safely carried by a thrust bearing varies with the velocity of the rubbing surfaces. The accompanying table may be used as a guide in designing bearings in which the shaft is made from wrought iron or steel and the bearing from bronze or

brass, and which have ample lubrication. In general, it is possible to use bath lubrication for thrust bearings, that is, the running surfaces are submerged constantly in a bath of oil. If the shaft is made from cast iron running on bronze or brass bearings, the values in the table for allowable pressure should be only one-half of those given. In designs where the motion is slow and heating cannot well result, as in pivots for swing bridges and similar constructions, pressures up to 4000 pounds per square inch are permissible.

Allowable Pressure, in Pounds per Square Inch, on
Thrust Bearings

Average Velocity of Rubbing Surface, Feet per Minute	Safe Pressure, Pounds per Sq. In.	Average Velocity of Rubbing Surface, Feet per Minute	Safe Pressure, Pounds per Sq. In.
Slow and Intermittent	1500	100 to 150	75
50	200	150 to 200	60
50 to 100	100	Over 200	50

Collar Thrust Bearings. In collar thrust bearings, the thrust is taken by projections or shoulders on the shaft, often at some distance from its end. This type of bearing is used when a greater thrust than can be conveniently placed on a single flat or step bearing is to be taken care of. In a well-made bearing, each of the collar surfaces takes its proportionate part of the load, and it is thus possible, without using excessive diameters, to distribute properly a very great thrust on a number of collars formed solidly with the shaft by cutting a number of grooves in the latter. One advantage of the collar bearing is that the difference between the outer and inner diameters of the bearing surface is not very great, and hence the velocities at the outer and inner edges do not vary appreciably; this, again, eliminates unequal wear on the thrust collar surfaces. The safe load that may be placed on collar thrust bearings varies between 60 to 100 pounds per square inch.

Collar thrust bearings are commonly designed with special thrust washers let into the bearing proper, these thrust

washers bearing against the collars on the shaft. The outer diameters of the collars are usually made not more than one and one-half times the diameter of the shaft. In the case of small bearings of cheaper design, the bearing surfaces of the thrust bearings are sometimes made integral with the bearing itself, but this design is not to be recommended, as it is difficult to distribute the load evenly between the different collars. When the main casting is made, say, of cast iron, and bearing washers of brass are inserted, it is possible to scrape each of these washers until the shaft collars bear properly against each washer, so that the thrust is uniformly distributed. In some cases, the bearing washers are made in the shape of a horseshoe fitted over the shaft, so that each washer can be removed without disturbing the bearing or the shaft.

Hydraulically Supported Step Bearings. The type of thrust bearing which is hydraulically supported has very little frictional resistance and is adapted to heavy pressures and high speeds. Bearings of this type which have been applied to Curtis vertical steam turbines are so designed that oil (water may also be used with this type of bearing) under sufficient pressure to sustain the load, is forced between the recessed plates at the bottom of the shaft and then passes out radially in the form of a thin film and up through a cylindrical guide bearing located just above the bottom plates, thus floating the shaft upon the oil film.

Thrust Bearing Design Based on Principle of Wedge-shaped Oil Film. The investigations of Professor Osborne Reynolds, following the experiments of Tower on well-fitted car journals and brasses flooded with oil, showed that the oil, because of its viscosity and adhesion to the journal, is, by the journal rotation, dragged into a wedge-shaped space between the journal and brass. This action sets up pressure in the oil film which, in turn, supports the load, thus separating the bearing surfaces. The design of the Kingsbury thrust bearing is based on this principle, the bearing floating the load on wedge-shaped oil films which

form automatically and without employing a high pressure oil pump. There is usually a flat annular revolving plate with the bearing face immersed in oil and supported on one or more shoes which are mounted to tilt as required by running conditions. These bearings are made for both horizontal and vertical shafts. The low-speed bearings may be loaded to 1000 pounds or more per square inch when using heavy oil and high-speed bearings with light oils regularly carry loads up to 500 pounds per square inch. The friction loss in this bearing is very low. According to an approximate rule for vertical bearings having six shoes with the inside diameter one-half the outside diameter and loaded to 350 pounds per square inch of shoe area, the mean coefficient of friction is 0.00009 times the square root of the revolutions per minute and varies inversely as the square root of the unit pressure, when using dynamo oil having a temperature of about 40 degrees C. The coefficient of friction has been found by a large number of tests to vary between 0.0008 and 0.003.

CHAPTER II

BEARING METALS

BEARINGS are usually composed of alloys of copper, lead, tin, antimony, and zinc, and are known as babbitt metal (after the name of the discoverer of this material), white metal, brass, phosphor-bronze, and various other trade names. Quite a number of these are patented, such as "plastic bronze," etc., but many are sold merely under trade names, and, in some instances, are of uncertain composition. The combinations of the metals enumerated, that are used for bearing purposes, may be grouped under the two heads of white metal and bronze. *Bronze* is the term which was originally applied to alloys of copper and tin as distinguished from the brasses, or alloys of copper and zinc; but gradually this term has become applied to nearly all copper alloys containing not only tin, but lead, zinc, etc., and no sharp lines of demarcation exist between the two. Thus white metals are made up of various combinations of lead, antimony, tin, copper, and zinc, and may contain as few as two elements, or all five. Bronzes are made up of combinations of copper, tin, lead, and zinc, all of them containing copper and one or more of the other elements.

The essential characteristics to be considered in any alloy for bearings are composition, structure, friction, temperature of running, wear on bearing, wear on journal, compressive strength, and cost.

It is impossible to have one alloy reach perfection in all of these requirements, and so it is important to study the possible compositions and determine for what purpose each is adapted. It has been shown that a bearing should be made up of at least two structural elements, one hard constituent to support the load and one soft constituent to act

as a plastic support for the harder grains. Generally speaking, the harder the surfaces in contact, the lower the coefficient of friction, and the higher the pressure under which "seizure" takes place. Consequently, the harder the alloy the better. A hard, unyielding alloy for successful operation must, however, be in perfect adjustment, a state of affairs unattainable in the operation of rolling stock. For this reason, the lead-lined bearing was introduced, and the practice of lining bearings has now become almost universal.

Comparison between Hard and Soft Alloys for Bearings. While the harder the metals in contact the less the friction, there will also be the greater liability of heating, because of the lack of plasticity or ability to mold itself to conform to the shape of the journal. A hard, unyielding metal will cause the concentration of the load upon a few high spots, and so cause an abnormal pressure per square inch on such areas, and produce rapid abrasion and heating. The bronzes will operate with less heat than softer compositions, while the softer metals will wear longer than the harder metals. In the matter of wear of journals, however, the soft metals are more destructive. Particles of grit and steel seem to become imbedded in the softer metal, causing it to act upon the harder metal of the journal like a lap. High-priced compositions are used that have but little resistance to wear compared with cheaper compositions, and low-priced alloys are in service that are not cheap at any price. It is generally conceded that soft metal bearings cause a marked decrease in the life of the journal, and yet they have many marked advantages.

Principal Requirements of Bearing Metals. The principal qualities which a good bearing metal should have are good anti-frictional properties, so as to withstand heavy loads at high speed, without heating, and sufficient compressive strength so as not to be squeezed out of place under high pressure, or crack or break when subjected to sudden shocks. In addition to these, many other properties must be considered in a choice of bearing metals depending upon

the special purpose for which the material is to be utilized. Temperature variation is often an important factor, especially in refrigerating plants, and the coefficient of expansion should be considered to prevent undue binding, with consequent destruction of the bearing, and the possible variation in other properties, such as brittleness, ductility, etc., under various temperature conditions. In addition, many bearings must operate under conditions where they are subject to chemical action, whether that of brine or ammonia in refrigerating plants, or acids, alkalies, etc., in chemical establishments, and in dynamo and motor construction and operation, the electrical conductivity must be considered as well. This statement applies equally to all bearings incorporated in electrical machinery, where these must serve as electrical conductors, such as the bearings for the wheels in trolley cars, etc.

The chief properties which have been developed to a greater extent than others in machine design are those of friction elimination and resistance to compressive loads. Theoretically, all metals have the same friction, according to Thurston, and the value of the soft white alloys for bearings lies chiefly in their ready reduction to a smooth surface after any local impairment of the surface. Under these circumstances, the soft alloys flow or squeeze from the pressure, forming a larger area for the distribution of the pressure, thus diminishing its amount per unit of area. Further, the larger the area over which the pressure is extended, the less becomes the liability to overheating and consequent binding. Thus the frictional properties of a bearing are in inverse ratio to their compressive resistance, and invariably the best bearing alloys, from a high speed standpoint, are unsatisfactory for utilization in heavy machinery. The introduction of an iron or steel grid to form the base of the main bearing, and to be filled with much softer bearing metals than could ordinarily be used, or in some cases even graphite, is a step in the right direction, and presents possibilities of great importance in this field of machine development.

Metals Used in Bearing Alloys. Lead flows more easily under pressure than any of the common metals, hence it has the greatest anti-frictional properties. A number of metals exceed lead in this property, but their cost or some other factor render them unavailable. Lead is the cheapest of the metals, except iron; the comparative prices of the metals used in bearing alloys, under normal conditions (previous to the war), were about in the following order per one hundred pounds: Lead, \$4; zinc, \$5; antimony, \$9; copper, \$13; and tin, \$30 or more. It can thus be seen that the more lead that is used in a given bearing, the softer it is, the less friction it possesses, and the cheaper it can be furnished. It is, however, too soft to be used alone, as it cannot be retained in the recesses of the bearing even when used simply as a liner and run into a shell of brass, bronze, gun-metal, or some other alloy. Various other metals have been alloyed with it, such as tin, antimony, copper, zinc, iron, and a number of non-metallic compounds, such as sodium, phosphorus, carbon, etc., and the effect of the different ingredients is now fairly well understood.

Alloys Containing Antimony. If antimony is added to the lead, it increases its hardness and brittleness, and if tin is added as well, it makes a tougher alloy than lead or antimony alone. Nearly all of the various babbitt metals on the market are alloys of lead, tin, and antimony in various proportions, with or without other ingredients added. In such babbitts, the wear increases with the antimony and the price, with the tin. The higher antimony babbitts are used in heavy machinery, as they are harder, while those low in antimony are used in high-speed machinery. The steady increase in speed at which various operating units are maintained is responsible for a wide deficiency in this field in the duty performed by the bearing metal. The chief difficulty at present, in the operation of the modern turbine, is undoubtedly the maintenance of satisfactory bearing surfaces. Soft babbitts have never sufficient strength to sustain the weight and shock of heavy machinery bearings and can be used only as liners. The tendency to increase in

speed as well as in weight or size of machinery is limited simply by the satisfactory operation of the bearing metal itself.

Alloys of Lead and Antimony. Lead and antimony will alloy in any proportion. With an increase in antimony, the alloy becomes harder and more brittle. It has been determined that, when it is made of 13 parts of antimony and 87 parts of lead, the composition will be of homogeneous structure. If there is a greater proportion of antimony, free crystals of antimony will appear, imbedded in the composition; and, if less than 13 per cent, there appear to be grains of the mixture itself imbedded in the lead as the body substance. According to a theory generally accepted, an anti-frictional alloy should consist of hard grains, to carry the load, which are imbedded in a matrix of plastic material, to enable it to mold itself to the journal without undue heating. Such a condition would be met in a lead and antimony alloy having above 13 per cent antimony, but it is not advisable to use more than 25 per cent antimony, as the composition would be too brittle. The Pennsylvania Railroad Co. has adopted the 13 per cent antimony-lead alloy as a filling metal for bearings, in order to obtain the best results.

The friction becomes less with an increase of antimony, and the temperature of running is likewise diminished when running under normal conditions; but the harder the alloy, the more difficulty is experienced in bringing it primarily to a perfect bearing, and the greater the liability of heating through aggravated conditions. The wear on the journal is not decreased with increasing hardness, as might be expected. This journal wear is in all probability not due so much to the alloy directly as it is to the fact that the softer metals collect grit, principally from the small particles of steel from the worn journal, and, acting as a lap, cause rapid wear. With the harder metals these particles are worked out without becoming imbedded. The cost of the lead and antimony alloy is very low. It can be used in many services where higher-priced alloys are relied upon mainly for their high cost. It is one of the greatest ex-

travagances of large industrial establishments to use materials that are too good for certain uses, and even perhaps unsuited, under the supposition that they must be good because they are expensive. This fact has no greater exemplification than in the purchase of babbitt metal, and is due to the great uncertainty which exists not only among users, but among the manufacturers of these bearing metals.

Alloys of Lead, Antimony, and Tin. It should not be assumed that antimony-lead is the cheapest alloy to use under all circumstances, because, when high pressures are to be encountered, tin is a very desirable adjunct. Tin imparts to the lead-antimony alloy rigidity and hardness without increasing brittleness, and can produce alloys of sufficient compressive strength for nearly all uses. The structure of a triple alloy of this nature is quite complicated, and not yet sufficiently defined. The cost of the alloy increases with an increase of tin; but, for certain uses, where sufficient compressive strength cannot be obtained by antimony, because of its accompanying brittleness, it is indispensable, and will answer in nearly every case where the tin basis babbitts are used.

Alloys of Tin and Antimony. These alloys are seldom used alone as bearing metals, but are extensively used for so-called "Britannia ware," and in equal proportions for valve seats, etc.

Alloys of Tin, Antimony and Copper. This combination is what is known as genuine babbitt, after its inventor, who presumably was the first man to conceive the idea of lining bearings with fusible metal. Alloys of this composition are among the most generally used bearing metals. They form a large group of varying compositions, some of which are given in the following, under "Babbitt Metal."

Alloys of Tin, Antimony, Lead, and Copper. Lead, although a soft metal, renders this alloy, when added in but small proportions, harder, stiffer, more easily melted, and superior in every way to the alloy without it. This is one of the instances where cheapening of the product is beneficial.

The foregoing represents the more important combinations of alloys of tin and lead basis. These alloys are of far more importance in the arts than the white metals, the main portion or basis of which is zinc. At various times new combinations of zinc have been proposed, but, with very few exceptions, they have not come into common use for two reasons: First, because of the great tendency of zinc to adhere to iron when even slightly heated. What is technically known as galvanizing the journal is caused by these conditions; second, because of the brittleness produced under the effects of heat, such as is produced by friction

Table I. Babbitt Metal Compositions

Class of Service Adapted for	Composition of Metal			
	Tin	Anti-mony	Copper	Lead
High-pressure bearings.....	90	7	3	...
High pressure and fast speed.....	86	12	2	...
Medium pressure and high speed..	30	20	50
Medium pressure and medium speed..	15	25	60
Low pressure and medium speed...	8	20	72
Principally for shaftings, etc.....	10	90

when lubrication is interfered with, and consequent danger of breakage.

Babbitt Metal. Babbitt is the name given to a great variety of white-metal alloys used as linings for bearings. The name is derived from that of the inventor, Isaac Babbitt, who, in 1839, obtained a patent for a special type of bearing enclosing a soft-metal alloy. This bearing had lips extending around the ends to retain the soft metal in case of accidental heating, and also to prevent the soft metal lining from being spread out when subjected to heavy pressure.

Composition of Babbitt Metal. The exact composition of the original babbitt metal is not known. The ingredients were copper, tin, and antimony, in approximately the following proportions: 89.3 per cent of tin; 3.6 per cent of copper; 7.1 per cent of antimony. This metal possesses great anti-frictional qualities, but the high percentage of tin makes it expensive and has led to the substitution of

other metals which are marketed under the name of "babbitt metal." These cheaper grades, when properly made, are superior to the original babbitt metal for some purposes. The composition of babbitt metal should be varied according to the pressure to which it will be subjected and the speed of the rotating member; the size of the bearing and thickness of the babbitt metal lining should also be considered. While it is not necessary to use a different composition for each slight variation, a different grade is preferable when the conditions are radically different. The compositions of metals for different classes of bearings frequently used are given in Table I.

Table II. Babbitt Metal Compositions

Number	Tin, Per cent	Antimony, Per cent	Copper, Per cent	Lead, Per cent
1	83.33	8.33	8.33
2	89.00	7.00	4.00
3	50.00	15.00	2.00	33.00
4	5.00	15.00	80.00
5	10.00	90.00

Babbitt for Sub-press Slides. One of the uses for babbitt is in sub-presses where the plunger slides up and down. A special babbitt is used for this purpose consisting of 66 per cent of lead, 18 per cent of antimony, and 16 per cent of tin.

American Society for Testing Materials Specifications for Babbitt Metals. A sub-committee of the American Society for Testing Materials has proposed to reduce the large number of babbitt metals in use to five, a number which the committee thinks will be ample for every class of work. Table II gives the composition of the series which it is believed covers the range for all requirements.

Where cost is of no great importance, the original babbitt formula is still considered the standard of excellence in the trade, and has been adopted by many of the leading railroads, the United States Government, and many industrial establishments. It is used in the majority of cases where cheaper composition would do equally as well. It is the most costly bearing alloy, due to the high content of tin.

S. A. E. Standard Babbitt Metal. The babbitt metal adopted as a standard by the Society of Automotive Engineers, Inc., is a special grade owing to the large amount of copper contained therein. It is used for the connecting-rod linings of motor bearings, or any service where machinery designers are confronted with severe operating conditions. The composition follows:

Tin	84.00 per cent
Antimony	9.00 per cent
Copper	7.00 per cent

A variation of 1 per cent either way will be permissible in the tin, and 0.5 per cent either way will be permissible in the antimony and copper. The use of other than virgin metals is prohibited. No impurity will be permitted other than lead, and that not in excess of 0.25 per cent.

Properties of Babbitt Metal. Babbitt metal and the white-metal alloys, generally, not only have valuable "anti-frictional" properties, but other important advantages. They may easily be melted in an ordinary iron ladle, so that little equipment is required for re-lining a bearing; they are durable and wear remarkably well; they tend to reduce shocks and deaden noise; and they can readily be provided with grooves for lubrication and are easily fitted to obtain a uniform bearing. These alloys, however, are liable to melt and run out of the bearing shell in case of accidental overheating. Overheating of other bearing materials which would not melt and flow, however, would cause equally disastrous results as to their bearing properties. The following information on babbitt bearing metals was given by T. J. Johnston in *The Electric Journal*.

A good bearing material must fulfill the following requirements: It must be of sufficient strength to sustain its load; it must not heat rapidly; it must be easily worked; it must have good anti-frictional properties; it must have a long life with small loss of material due to wear; and (with the exception of cast iron on cast iron and hardened steel on hardened steel) it will usually be a material of an entirely different molecular construction from that of the revolving journal which it must support. The journal, when running,

may be completely borne by the oil film but, during the time of starting or stopping, the film is broken, minute irregularities on the surfaces of the bearing and journal engage, and, if the bearing does not yield, as in the case of a steel bearing and a steel journal, small particles are fused and torn out; these accumulate at the entrance point, and may cut both the bearing and the journal. With a steel journal running in a white-metal bearing, the bearing surface is entirely different in its molecular structure, the bearing inequalities are not strong enough to resist the minute inequalities in the journal, and so, instead of fusing, they yield and are smoothed out; consequently, the bearing surface, instead of being injured by contact and momentary high coefficient of friction, is smoothed and burnished, thus preparing the way for a uniform wedge oil film with a minimum coefficient of running friction.

Laboratory tests show that a lead-base babbitt will give very good results, but such tests of bearing materials are difficult and uncertain, and apt to be misleading. Similarly, chemical tests cannot be relied upon entirely, since a great deal is dependent on the actual making of the babbitt. Laboratory and chemical tests, taken in conjunction with actual service tests, furnish reliable data, and a bearing metal developed along these lines may be counted upon to give consistent results in practical work. Babbitts made up according to nearly 300 different formulas are at present on the market. It would be of very great benefit to the users of babbitt if this number were greatly reduced and the process of manufacture so standardized as to insure a uniform quality of alloy. Except for a few cases, but two babbitts, one a lead-base alloy, the other a tin-base alloy, each being the best that can be made, are required for a complete line of bearings, ranging in weight from a few ounces to several tons.

Bronze Bearing Metals. Bronze is the term which originally was applied to alloys of copper and tin as distinguished from alloys of copper and zinc. A bronze is usually understood to have more copper than tin, and the properties of the metal differ widely according to the percentages of these

constituents which are present. In general, the alloy hardens when tin is present up to proportions of 30 per cent or a little over, and, when this limit is exceeded, it takes on more and more the nature of tin until pure tin is reached. From a scientific point of view, this alloy is one of the most interesting, and has attracted the attention of many investigators, who have spent years of study on it, to learn its various properties and explain its constitution.

The alloys of interest in this connection, however, are those which are so constituted as to be adapted for bearing purposes. These would be said to contain from 3 to 15 per cent of tin, and from 85 to 97 per cent of copper. The alloy of tin containing a small percentage of copper is often used as a babbitt metal, but this comes under the class of white metals, which have already been discussed. Bronze containing above 15 per cent of tin has been recommended at various times for bearings, owing to its hardness, but such a bearing demands mechanical perfection and perfect lubrication. It has no plasticity of its own, and, as soon as the oil film is interrupted, rapid abrasion and "seizure" take place, with hot boxes as the result. The very erroneous idea is still held by many that, to resist wear and run with the least possible friction, a bearing alloy must be as hard as possible. It is true that hard bodies in contact move with less friction than soft ones, but the alloy which is the least liable to heat and cause trouble is the one which will stand the greatest amount of abuse; that is, an alloy which has sufficient plasticity to adapt itself to the irregularities of service without undue wear. The alloys of copper and tin were used extensively some twenty or twenty-five years ago, and were considered the standard for railroad and machinery bearings. The old alloy, known as "cannon bronze," containing 7 parts of copper and 1 part of tin, is still specified by a few unprogressive railroad men and machinery builders.

Bronze Containing Copper, Tin, and Lead. This composition is now the recognized standard bearing bronze, its advantage over the bi-compound coming from the introduction of lead. The bronze containing lead is less liable to heat

under the same state of lubrication, etc., and the rate of wear is much diminished. As lead is cheaper than tin, it is desirable to produce a bearing metal with as much lead and as little tin as possible. The metal known as "Ex. B." composition (tin, 7 per cent; lead, 15 per cent; copper, 78 per cent) is stated to be the best that can be devised. This alloy contains the smallest quantity of tin that will hold the lead alloyed with the copper. By adding a small percentage of nickel, however, to the extent of from $\frac{1}{2}$ to 1 per cent, a larger proportion of lead may be used, and successful bronzes have been made by this process, which contained as much as 30 per cent of lead. Such bronzes, containing a large amount of lead, through the addition of nickel, are known in the trade as "Plastic bronzes" and are a regular commercial article.

Undoubtedly, in investigations in this field, sufficient attention has not been paid to the effect of temperature on the bearing properties of the alloys used for these bearings. More rigid investigation in this field and limitations in regard to the temperatures permissible, with means for maintaining these within fairly close limits, will undoubtedly result in a great increase in the possibility of improvements in speed and weight of various types of machinery. More or less extensive experiments along these lines are being conducted in regard to the bearings used in turbine construction, since the speed here has rendered the problem an acute one and is necessary for efficient operation of the turbine itself.

Commercial Bearing Metals. Table III, "Composition of Alloys Used for Bearing Metals," shows the various constituents of the more or less common bearing metals now on the market. A wide deviation in the composition of babbitt is shown in the first part of the table. The first babbitt is a fairly good alloy for high-speed machinery, but is not very hard. Its melting point is about 500 degrees F.; in fact, the properties of all alloys or bearing metals can be very widely deduced from their melting point. The second babbitt is somewhat harder and melts at a higher point. Both of these are used largely for lining purposes. The

fourth babbitt is used very widely for heavy machinery. All of the babbitts mentioned have been fairly successful. Babbitt 6 has good wearing properties, but cannot be used for high speeds. Most of the other metals included in the table, where copper is not used in excess, can be regarded as in the same class as babbitts. The "white" class has a fairly good electrical conductivity, much greater than that of ordinary babbitt, and is used in the bearings of generators, motors, electric cars, etc. In alloys containing sodium

Table III. Composition of Alloys Used for Bearing Metals

Alloys	Lead	Tin	Anti- mony	Cop- per	Zinc	Other Constit- uents
Babbitt 1.....	80.0	20.0
Babbitt 2.....	72.0	21.0	7.0
Babbitt 3.....	70.0	10.0	20.0
Babbitt 4.....	80.5	11.5	7.5	0.5
Babbitt 5.....	0.5	68.0	1.0	30.5
Babbitt 6.....	20.0	80.0
Babbitt 7.....	86.0	10.0	4.0
White metal 1....	82.0	12.0	6.0
White metal 2....	3.0	7.6	3.8	2.3	83.3
White brass.....	64.0	2.0	34.0
Bronze 1.....	11.5	11.5	77.0
Bronze 2.....	15.0	8.0	76.2	P = 0.80
Bronze 3.....	9.5	10.0	80.5
Bronze 4.....	3.0	97.0
Bronze 5.....	15.0	85.0
Bronze 6.....	15.0	7.0	78.0

Note: P indicates phosphorus.

the oxidation of the sodium produces a material which will saponify with the oil used in the bearing and produce soap, thus assisting lubrication. Practically no experiments have been made to determine the extent and amount of such action. Possibilities along this line, however, are great, not only for this particular alloy, but for many others not as yet considered.

The other alloys included in the table consist, to a very great extent, of copper, tin, and lead, and usually have a thin liner of lead or some soft babbitt, and hence wear much better than an entire bearing of the soft babbitt. The tendency to wear decreases with increase of lead and increase

of tin. Increase of lead, of course, affects the frictional quantities of the alloy, hence its heating properties. A certain amount of other metal, however, is necessary to keep the lead from separating from the copper. The "P. R. R. car brass, B" is considered one of the best bearing bronzes that can be obtained. It contains approximately the smallest quantity of tin that will hold the lead alloyed with the copper. Table IV, "Composition of Bronzes," gives a list of alloys used by the U. S. Navy Department.

Table IV. Composition of Bronzes

Alloys	Lead	Tin	Antimony	Copper	Zinc
White metal.....	3.0	7.6	3.8	2.3	83.3
Hard bronze for piston rings	22.0	78.0
Bearings — wearing surfaces, etc..	13.5	83.0	3.5
Naval brass.....	1.0	62.0	37.0
Brazing metal.....	85.0	15.0

Substitutes for Tin in Bearing Metals. The Bureau of Standards, in its efforts to determine to what extent substitutes may be used for tin, calls attention to the limited supply and great demand for this metal, and mentions that it is imperative that steps be taken at once to eliminate all waste of tin and to use substitutes or reduce its use as far as possible and practicable in all alloys. It is pointed out that many specifications for bearing metals now in existence call for pure tin, and that a large saving of high grades of tin, such as Banca or Straits, could be brought about by allowing the use of second quality pig tin in making tin-base babbitt. Detrimental impurities could still be limited, but a maximum of 1 per cent of lead could be allowed. This would not be harmful to a tin-base or lead-base lining metal. There is no question but that the tin content could be reduced somewhat in all bearing alloys. Every possible saving should be effected. For those cases where genuine babbitt is now used and which require a very high grade of lining, alloys are suggested containing either of the combinations,

85 per cent tin, 10 per cent antimony, and 5 per cent copper; or 65 per cent tin, from 3 to 6 per cent copper, and from 28 to 30 per cent zinc.

Lead-base linings can be satisfactorily used in many cases where tin-base linings are now in use, and in fact the change has already been made in some shops, although the substitution should be more general. Several special types of lead-base linings, hardened with alkali earth, are reported to be giving very satisfactory service in the place of high tin babbitt. Two other alloys containing large percentages of lead and zinc have apparently been found to perform the same services which were required of tin-base linings in machine tool work. One of these alloys consists of 8 per cent tin, 8 per cent antimony, 4 per cent copper, and 80 per cent lead; and the other consists of 5 per cent tin, 7 per cent antimony, 2 per cent copper, 10 per cent lead, and 76 per cent zinc.

One way in which a lining metal can be saved is to use as thin a lining as is possible in order to maintain a high enough temperature during pouring to guarantee a firm bond and solid mass of metal. In place of a bronze bearing containing 80 per cent copper, 10 per cent tin, and 10 per cent lead, an addition of a small percentage of zinc or phosphor-copper and an increase of the lead content will result in a saving of at least 25 per cent of the tin ingredient.

Comparison of Lead-base and Tin-base Babbitt Metals. Tenacity is desirable in a bearing metal, especially at the higher bearing temperatures, as bearings fail because of warping or deformation. The following information on the relative merits of lead-base and tin-base babbitt metals was abstracted from a paper presented before the American Institute of Mining Engineers by J. L. Jones, metallurgist, Westinghouse Electric & Mfg. Co.

The Brinell test is commonly regarded as a measure of tenacity; in fact, the proposition has been made to substitute for the term "Brinell hardness number" the expression "tenacity number." Brinell tests at progressively increasing temperatures showed that the lead-base babbitt has a better resistance to deformation at the working temperatures than babbitts with a tin base. The tests were made on disks 4

inches in diameter and $1\frac{1}{2}$ inches thick of the following composition:

Babbitt	Antimony, Per cent	Copper, Per Cent	Lead, Per Cent	Tin, Per Cent
A.....	8	2	0	90
B.....	8 $\frac{1}{3}$	8 $\frac{1}{3}$	0	83 $\frac{1}{3}$
C.....	14	0	78	8

The disks were heated by an electric hot plate, the temperature being controlled by suitable rheostats. Pyrometer leads were soldered in the center of each disk. The disks were well insulated to prevent radiation loss and were held at the desired temperature for several minutes to guard against variation. The tests were made on the bottom surface of the disks after a light machine cut was taken to secure a perfectly planed surface. The Brinell hardness numbers obtained at the various temperatures were plotted and the results are shown by the curves given in Fig. 1. At 35 degrees C. the hardness of the babbitts A and C is identical, but above this temperature the lead-base babbitt has the higher hardness number. The curves of babbitts B and C are almost parallel and not very far apart. Complete results from various test floors, covering a number of gears and a variety of motors, confirm the superiority of the lead-base babbitt. In one case where it was necessary to reline one hundred bearings containing babbitt A in a month, it was necessary to reline only about six bearings that contained the lead-base babbitt C. As a result, the lead-base babbitt was substituted for all classes of machines and the tin-base babbitt A eliminated altogether.

While the Brinell hardness shown in the chart for the babbitts A and C is not far from the average hardness found for these alloys when using the standard hardness test piece, the results obtained for the hard babbitt B is much below the normal. This probably is due to the difficulty of preventing the large amount of copper in this babbitt from segregating even when kept very hot and being stirred continuously. The copper falls to the bottom of the melting pot; hence when stirring, the aim should be to bring the metal from the bottom of the pot to the top.

Effect of Lead on Babbitt. It is a common belief that the addition of even a small amount of lead to a genuine babbitt renders it inferior. Fig. 2 shows the results of tests made with the tin-base babbitt A to which has been added 1, 3, and 5 per cent of lead, the results being shown by curves B, C, and D, respectively. These results show that when a

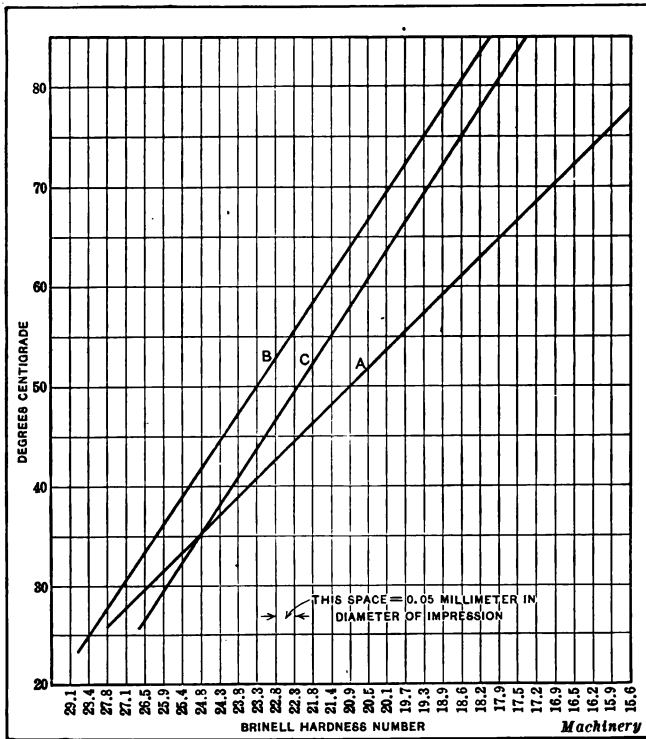


Fig. 1. Hardness of Representative Babbitts at Varying Temperatures

small amount of lead is accidentally added to the tin-base babbitt its hardness and anti-frictional qualities are much improved.

Effect of Compression on Brinell Hardness of Babbitt. In the case of large bearings, peening or compressing the babbitt by means of hammering is often specified, it being supposed that by just compressing or densifying the babbitt

and hardening it the bearing will give better service. In one case where two phosphor-bronze plates were coated with babbitts B and C, then subjected to pressures varying from 8500 to 13,000 pounds per square inch, it was found that the lead-base babbitt stood up better than the tin-base babbitt. When the load was increased to 30,000 pounds per square

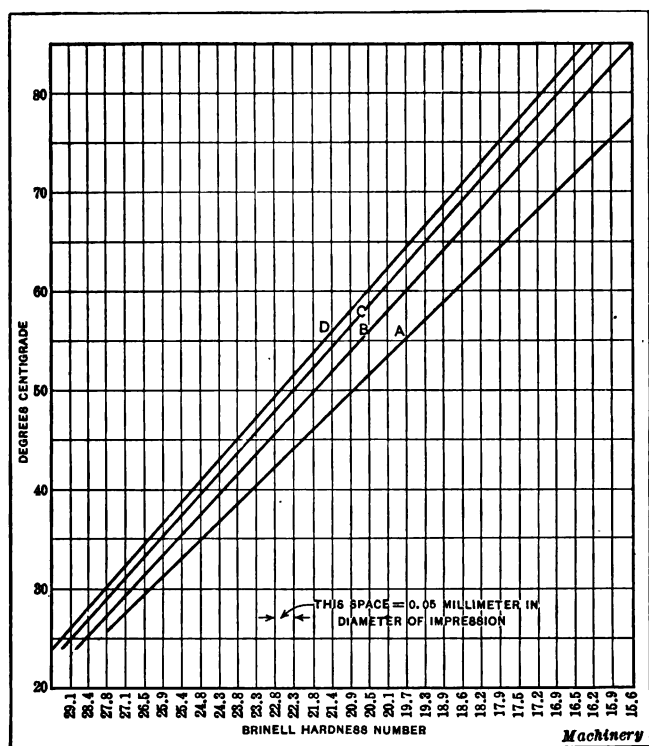


Fig. 2. Effect of Lead on Hardness of a Babbitt

inch, the tin-base babbitt presented a better appearance, as it flowed uniformly over the edge of the bronze square in all directions, while the lead-base babbitt was compressed more on one side than on the other. The tests show that broaching, peening, etc., do not appreciably increase the hardness of babbitt; hardness must be obtained through quickly cooling the babbitt lining by means of water-cooled mandrels, etc. A microscopic examination of a lead-base babbitt

shows that the metals tend to segregate. This lack of uniformity may be guarded against by pouring a thin lining and chilling quickly. The secret of obtaining good bearings consists in keeping the matrix tough and hard. There is less tendency for tin antimonide crystals in tin-base babbitts to rise to the surface, because of the lower gravity of these babbitts.

Miscellaneous Bearing Metals. A high-class bearing metal is prepared as follows: Melt 7 parts of copper at as low a heat as possible; then add 25 parts of antimony and 200 parts of tin. This mixture is cast in iron ingot molds. It is then remelted and to each five pounds of the ingots is added eight pounds of tin, this second alloy being cast in bars to suit the requirements.

For an anti-friction metal that can be subjected to pressures up to about 400 pounds per square inch, the following composition has been recommended: Lead, 85 per cent; antimony, 10 per cent; and tin, 5 per cent. For pressures exceeding 400 pounds per square inch, the following alloy will prove satisfactory: Tin, 85 per cent; copper, 5 per cent; and antimony, 10 per cent. This alloy can be used safely for pressures up to 1000 pounds per square inch. The alloys in Table V are stated to have given complete satisfaction for the purposes mentioned.

An alloy made by melting together approximately equal parts of cadmium and zinc, with an addition of a small proportion of antimony is said to be superior to the usual white metal. It is very easily worked and particularly easily turned, it fills up the mold completely when cast, possesses relatively great hardness, and, what is most important, it has an extremely small coefficient of friction. The alloy can consist of from 45 to 50 parts of cadmium; from 45 to 50 parts of zinc; and up to 10 parts of antimony. The antimony added should not exceed 10 per cent, as otherwise the metal is too brittle. A very suitable proportion of antimony is 5 per cent. If the proportion of cadmium and zinc is considerably varied, the coefficient of friction increases, and the other good properties of the alloy are essentially prejudiced.

Tests on Bearing Metals. The R. K. LeBlond Machine Tool Co., Cincinnati, Ohio, has made a comparative test of bearings for lathe spindles on two of the engine lathes in its plant, to determine the relative durability of the following combinations, viz.: hardened steel journal in cast-iron box; hardened steel journal in bronze; soft steel journal in bronze; and soft steel journal in babbitt. The experiment was made on both ends of the spindles, thus making the four combinations named. Both lathes were kept in constant use, the general character of work being the same for both. When examined, the condition of the hardened steel journal and cast-iron box was the best of all, neither the spindle nor

Table V. Miscellaneous Bearing Metals

Used for	Tin	Lead	Zinc	Anti- mony	Cop- per	Bis- muth
Dynamos—high-speed.....	88	8	3.5	0.5
Marine engines.....	77	17	...	3	3
Eccentrics.....	5	77.75	...	15	2	0.25
Submerged bearings.....	40	48	...	10	2
Main bearings.....	34	44	...	16	6
Slides, thrust bearings.....	65	...	30	2.5	2.5
Railway trucks.....	42	...	56	...	2
Axle-boxes.....	74.55	13.50	1.80	6.55	3.6
Plastic metal.....	80	10	...	1	8	1
Genuine babbitt (hard).....	80	10	10	...
Genuine babbitt (No. 2)....	83	9	8	...
Universal bearing metal.....	6	77.75	...	16	...	0.25

the box being appreciably worn, the grinder and scraper markings still being visible. The hardened steel journal and bronze box combination was in good shape, but the journal was slightly ridged in the center, showing more wear than the first. The soft spindle in bronze was worn appreciably, but the soft spindle in babbitt was in first-class condition. The front bearings were provided with oil rings and oil reservoirs, and the main bearings with oil reservoirs and felt wicks.

Investigations to ascertain to what extent repeated meltings of bearing metals influence their strength and reliability showed the following results: As regards white metal (alloys of copper, antimony, and tin) it was found that re-

peated meltings did not noticeably alter the grain, but that the rate of cooling had a considerable influence. Quick cooling yielded a finer grain and a higher hardness and strength, and it is recommended that white metals should not be heated to high temperatures and that they should be cooled rapidly. Bronze, poor in tin, and, therefore, comparatively inexpensive, may have the hardness and strength increased by being rapidly cooled from a temperature of 1440 degrees F.

While a great deal of attention has been given to the investigation of the properties of bearing metals, there is still a great deal to be done, and many important points are yet to be determined. Apparently many bearing metals are unnecessarily expensive, containing higher percentages of high-priced metals than is required.

CHAPTER III

METHODS OF LUBRICATING BEARINGS

ASIDE from the selection of the lubricating oil, the proper lubrication of bearings requires a careful study of the best means of conducting the oil to the bearing surfaces and also means of protecting the bearing against the entrance of foreign material such as would injure the surface. Far more attention is now paid to these details than was the case formerly, especially in metal working machinery. The higher speeds and duties which prevail, the use of geared drives, and the practice of boxing-in portions of mechanism, have all had their effect upon the methods of lubrication, and many new and improved devices have been developed and come into general use. The tendency is always to render the lubrication, as far as possible, automatic in the most complete systems, so as to obviate frequent attention on the part of the attendant, and to minimize the consequences of neglect. The details of construction of machine tools are so complex and varied that it is not surprising to find a multitude of ways of supplying oil to the different elements, and the principal of these will be described in detail, with the help of typical illustrations. The question of lubrication divides itself naturally into two main heads, viz., supply and distribution. Subsidiary to these are, the prevention of loss before the oil has done its work, the prevention of access of dirt to the surfaces, and the final catching of the lubricant after it has left the surfaces, with or without immediate return. In many instances, it is necessary to prevent escape of oil to belts and surfaces where its presence is undesirable.

Oil Supply. The oil supply is effected by gravity, by pressure, by capillary attraction, by a mechanical lifting action which raises oil from a well, or by contact, through the

medium of wicks, pads, or rollers. The distribution is effected by rows of holes, or grooves, by wicks or pads laid suitably, by rollers, or by the splash method. Such varied conditions are met with that it is impossible to claim any particular mode of lubrication as being the best for shafts or slides. The speed of rotation may make a vital difference in the results, and a method of oiling that is quite satisfactory in one machine may be undesirable or impracticable in another, on account of inaccessibility of the parts or great increase of pressures. With the use of the boxed-in geared drives in so many types of metal working machines, the difficulties of reaching the parts have increased, and some rather elaborate devices have been evolved for supplying and catching the oil without having recourse to the removal of covers. Pipes naturally play an important part in the conduction, and there is often extensive drilling of shafts and bearings for the purpose of conveyance from the outside.

Quantity of Oil. One of the greatest differences which affect the lubricating problem is the quantity of oil which has to be supplied, and this depends on the function of the moving part. If it runs at a good speed, under considerable duty, a copious supply is essential to prevent cutting and heating, but if the movement is slow or intermittent and the work light, a simpler system meets the case. In sliding ways, which run at high speeds and carry much weight, ample lubrication is imperative, while, on the other hand, such parts as cross-slides, tool-boxes, etc., which move slowly or intermittently will retain their film of oil for a long period without a new supply. The horizontal or vertical dispositions of sliding surfaces also make some difference in the amount of oil which can be fed. Certain horizontal ways admit of flooding without inconvenience, while this cannot be done with a vertical slide, nor would it usually be necessary. Even when ample lubricant is fed to a bearing or slide, this is of little avail unless distributed correctly and evenly, so that no parts of the surfaces become dry.

It is not sufficient to drill a hole and trust that the oil will get between two surfaces beyond the vicinity of the hole, although this is often done in cheap work. The oil must be distributed to the right and left in a definite manner, with due allowance for possible clogging. The two chief methods of effecting the spreading are grooves or channels, and pads of felt. The first named is satisfactory provided the quantity fed in is sufficient to carry it the required distance; the latter has the merit of absorbing a good deal of oil and keeping the surfaces moist for a certain period, even if neglected, and it also prevents the access of grit if properly fitted, which grooves alone do not. Sometimes a compromise is made, using grooves and pads together. If the pads are let into a well of oil, a certainty of supply is guaranteed, although this will not distribute such an amount of oil as some other methods.

The arrangements for catching and draining the oil from a surface or bearing influence the mode of feeding to a certain extent, and the two must be intimately related. A copious quantity of oil does not do much good if it is permitted to run out quickly and has to be collected in a primitive fashion, with inevitable waste. Under the pump and flooded bearing system, this is practicable, but, for ordinary use, a moderate amount supplied at intervals is better. When, however, each bearing is self-contained, with a ring-oiling or similar system, the amount of oil passing constantly is not restricted, but depends upon the method of lifting adopted. There is no waste here, because the oil from the surfaces returns to the reservoir and is carried again to the points at which it is needed.

Oil Distribution. In the means provided for distributing and catching the oil, widely varied methods are adopted, ranging from simple pots or troughs under open-ended bearings, to the highly-elaborated systems in which a forced supply is fed to the bearings, and is completely collected from these and returned to the pump for use over again. The latter system, if properly embodied in the design of a machine, is the best solution of the lubricating problem,

since the supply is constant and ample, and the trouble of attending to numerous lubricators is obviated. More thorough flushing of the bearings is also insured, and any dirt is carried away quickly, instead of being churned up for a considerable period, to the detriment of the surfaces, while the cooling is also more effectual. With the same idea in view it is often the practice to obtain the oil for the journal bearings of a gear-box from the well in which the gears splash around, in place of providing each bearing with a separate well.

Methods of Supplying Lubricants. The methods of feeding lubricants vary with the circumstances. The most important are:

1. The simple oil-hole, fed from a can, with no means of retaining a supply.
2. The oil-cup with forced feed, obtained by screwing down a cap or plunger.
3. The siphon cup with a wick feeding continually by capillary attraction.
4. The grease-cup device with constant feed by spring pressure.
5. The distributing box system with pipes and control taps to admit certain quantities of oil to the leading-out pipes.
6. The needle lubricator in which the feed is produced by the vibration caused by the rotation of the shaft.
7. The oil-well or reservoir which is filled to a definite depth, and serves to lubricate by wicks, etc., or by the splash method for a long period.
8. The force-pump which delivers a large amount of oil to one or several locations.

It is necessary to provide some way of observing the supply of oil. Many of the wells and lubricators may be inspected by opening a cover or door, or moving back a slide. Where, however, there is no means of readily accomplishing this inspection, sight or gage-glass tubes are fitted in which the height of lubricant is visible, or a glass window is attached to the wall of a box or well. Glass-bodied lubri-

cators often take the place of those with soiled metal bodies, in places where they are permissible. Occasionally a float gage is utilized to indicate the level of the oil. With a gear-box of ample capacity, no indicator may be necessary, since a certain amount is poured in, sufficient to last for so many weeks or months; and, in the case of some ball-bearings, a grease supply sufficient for a year or so is put in and the casing closed up.

The access of dirt and grit is guarded against by using strainers and filters. In a well sytem of lubrication the sediment naturally falls to the bottom, and care should be taken that it is not stirred up again; in fact, many bearings embody provision for keeping the sediment in such a way that it cannot possibly be returned. Drain-plugs at the lowest position provide for the drawing-off at intervals. When using felt pads, there is opportunity to make these assist in keeping the surfaces clean by a wiper-like action which prevents access of foreign particles between the rotating or sliding faces. The amount of protection which must be afforded to bearings and parts depends upon the nature of the processes carried on in the shop. Anything in the way of grinding demands rigorous guarding of all openings where access is possible, and felt pads and disks are used largely for such purposes.

Examples of Lubricating Devices. The most primitive mode of oiling is that of drilling a hole and pouring oil into it, leaving it to work its way along the bearing. An advance on this is to groove the bearing so that distribution takes place properly, and supply some means of insuring a constant flow of oil. A simple arrangement consists of a wick leading from the bearing to a small reservoir of oil. Such a bearing is very suitable for light and moderate duties, and will run for a long period without attention, although it is not so suitable for high-speed shafts which require flushing and continual movement of the oil to make them run cool. On many bushings and bearings the oil is not distributed by a single hole and groove in the journal, but instead there is a long slot and several holes lead from

it to the bearing surface. This is convenient, in some cases, especially where bearings are difficult of access, and where they have a longitudinal movement which would bring a single feeding opening out of the range of a pipe. Two holes are often drilled through near each end of a long bearing, so as to insure a distribution from both ends.

Thrust journals constructed with a number of collars may be lubricated on the separate passage system, Fig. 1, a hole for each collar leading from the common reservoir. This example is taken from a heavy planing machine table-

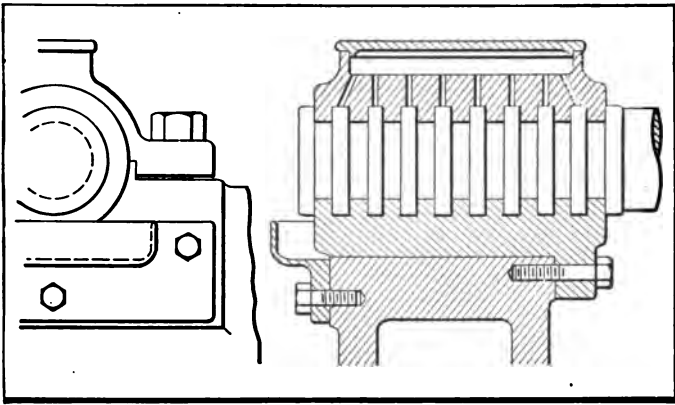
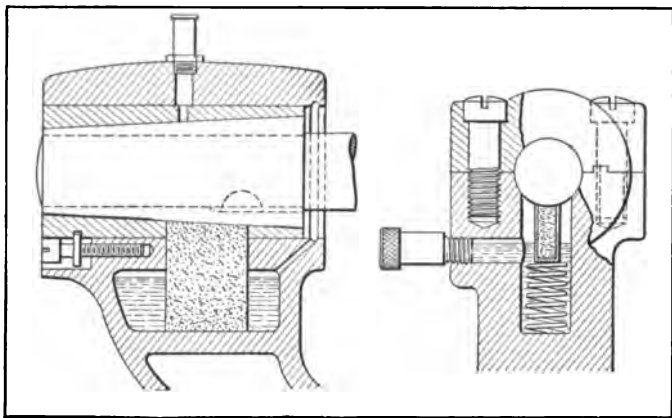


Fig. 1. Thrust Bearing with Oil Supply to Each Collar

driving screw; the illustration also shows the application of a trough below the end of the bearing to catch waste, a fitting that is very commonly used in various types of screws and other bearings that come at the ends of framings. Alternative practice is to cast a well below the collars of sufficient width to include them all, and fill this. The collars then are constantly smeared, avoiding the possibility of one or more running dry due to the hole being blocked with some foreign substance, as may happen with the arrangement shown in Fig. 1.

Lubrication by Felt Pads. Examples of various methods of applying felt pads for distribution are found in different machine tool designs, the felt pads being used either alone, or in combination with grooves in the bushing or on the

shaft. The felt not only insures a supply of oil on every part of the journal that it touches, but it filters the oil as well, and prevents the passing of grit or particles of metal. The pads are fitted into slots cut in boxes or bushings, and either dip into a well, or are simply fed through holes by a



Figs. 2 and 3. Different Applications of Pad and Reservoir Systems

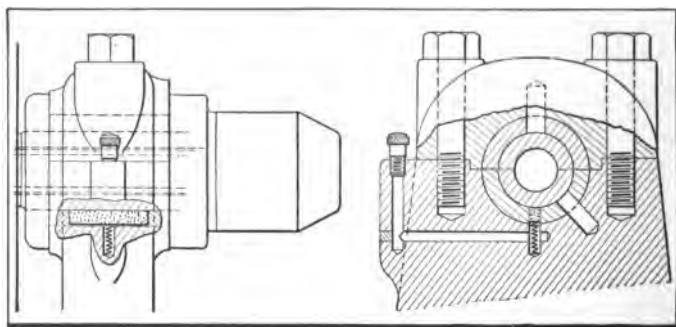


Fig. 4. Pad and Reservoir System with Supply from Side

cam, or from some type of lubricator. The pad is often let into a capacious well holding sufficient oil to last for a long period (Fig. 2) with provision for the return of the waste oil by a channel from the end or ends of the bearings. Figs. 3 and 4 show two variations in practice, the first having a side fed well, and a pad contained in a brass holder pressed

by a spring, and the second a modified arrangement, with the oiler set vertically. This disposition is sometimes more convenient for certain machines than placing it on top of the bearing cap, and there is the advantage that grit cannot reach the spindle, as it may when the supply is poured down from above. Alternatively to the fitting of a pad, wicking is sometimes adopted.

In Fig. 5 there is shown a form of construction which has been adopted for bearings used to support horizontal and vertical driving shafts on certain machine tools. Oil

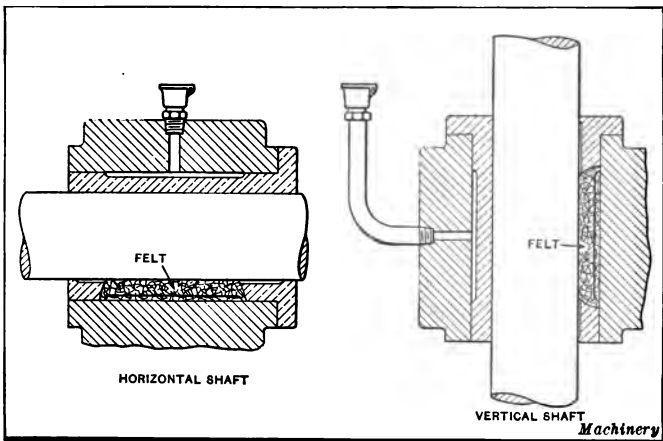


Fig 5. Method of Lubricating Horizontal and Vertical Machine Tool Driving Shafts

is delivered through a tube which carries it into an annular space machined in the outside of the bronze bearing box, and the oil flows around through this space to the opposite side of the box from the point at which the delivery tube is connected. Here there is a slot cut through the box to communicate with the inside or bearing surface, this slot being filled with felt. The oil flows through the felt to the bearing, and the felt serves the double purpose of filtering impurities from the oil and acting as a wiper which distributes the oil evenly over the surface of the bearing which it is required to lubricate. The illustration clearly shows the arrangement of this simple method of lubricating.

Ring Oiling. A system most extensively employed for spindles and shafts is the ring-oiling method, which insures a larger flow than is caused by the pad device. This method of providing for automatically delivering oil to a bearing requires one or more rings which are hung over the top of each journal and extend down sufficiently below the journal at the under side so that they dip into a reservoir filled with oil. As the journal rotates, it carries the rings around, thus bringing the portion of each ring which was formerly immersed in the oil up into contact with the top of the journal. The ring carries a considerable amount of oil

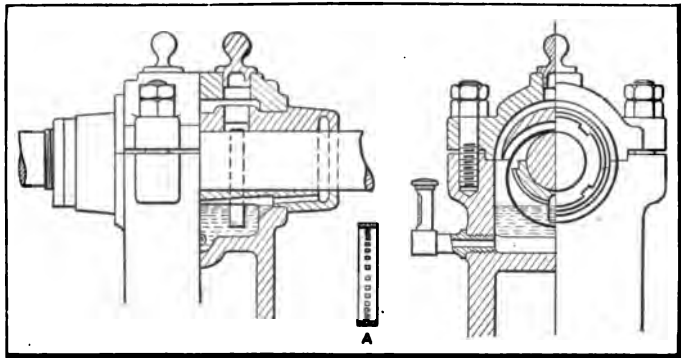


Fig. 6. Bearing with Two Oiling Rings and Height Gage
(A) Perforated Ring

with it, and in this way a constant supply of oil is deposited on the journal.

This method is used in conjunction with a reservoir for each bearing, or with a reservoir or box common to several bearings. If a gear-box, for example, has a body of oil into which the gears run, passages can be arranged to lead to the bearing wells, which simplifies arrangements; or the bearings may be constructed in a self-contained manner if they lie out of the plane of the oil box. In long journals it is often the practice to employ two oiling rings to provide greater quantity and better distribution of the lubricant. An instance of this is seen in Fig. 6, which also illustrates the gage-glass fitted to observe the height of oil. Sediment

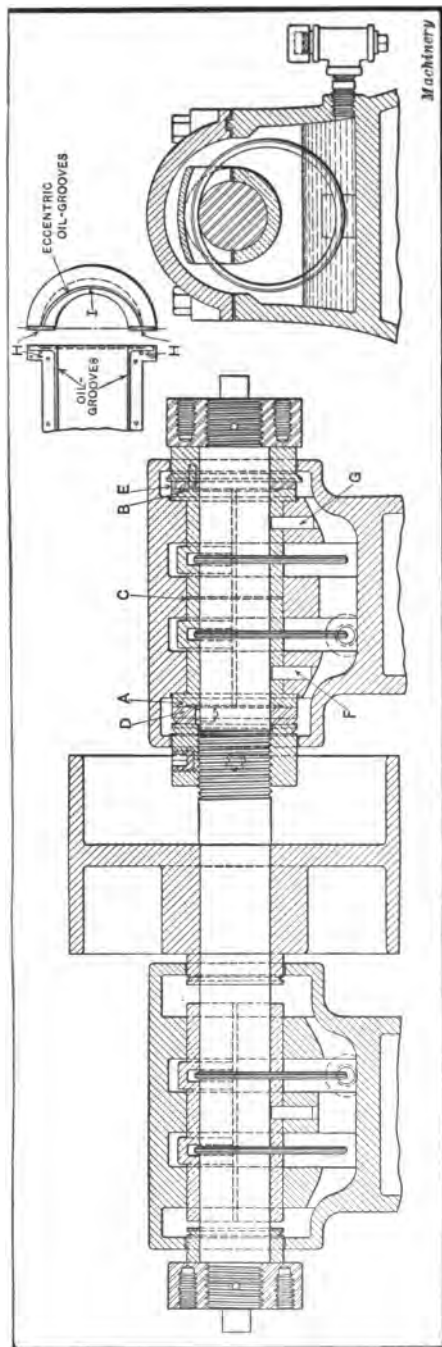


Fig. 7. Application of Ring-oiling Principle for Lubricating Combination Radial and Thrust Bearings on Disk Grinding Machine

collects in the well at the bottom, out of the range of the rings. In a few cases, the plain ring is not employed, because it will not lift enough oil for the requirements of the bearing, particularly if it should not happen to run freely in consequence of thickening of the oil. Some kind of auxiliary lifting is pressed into service, such as little scoops on the periphery, or projections,

or recesses, or the ring is punched with holes, as shown in Fig. 6, at A.

Ring-oiled Disk Grinder Bearings. A good example of the application of ring-oiled bearings in machine tool design is shown in Fig. 7, which illustrates the spindle of a disk grinder built by Charles H. Besly Co., Chicago, Ill. The bearings are so constructed that they are dustproof, which

is an important feature on the bearings of any grinding machine, and, in addition to the ring-oiled radial bearings, thrust bearings are provided to support the load which is applied in an endwise direction. This combination radial and double end thrust bearing is automatically oiled.

It is claimed that this ring-oiling device is so efficient in operation that the spindle literally floats on an oil film, thus greatly reducing the amount of power required to drive the machine. Tests which have been conducted to determine the transmission efficiency of these ring-oiling bearings are said to have shown that it is within 2 per cent of the efficiency of ball bearings. In the operation of ring-oiled bearings of this type, the oil should be drawn off about every six months, the exact length of time depending upon the conditions of service under which the bearings are operated. The oil wells have ample capacity; for instance, on the No. 17 Besly disk grinder, with a spindle $2\frac{1}{2}$ inches in diameter, the oil wells are 7 inches deep. Two solid steel rings are provided for each bearing, which operate through channels cut in the bearing bushings in such a way that it is impossible for a ring to be displaced and fail to operate properly. The bearing boxes are split to facilitate assembly, and the housings in which these boxes are inserted are so arranged that replacement of the boxes is an easy matter when this becomes necessary. A special grade of phosphor-bronze is used for the boxes, and they are designed with ample bearing surface so that there is very little wear.

On disk grinding machines, considerable end thrust is set up when work is forced against the grinding wheel, and this is particularly true in the case of machines furnished with a lever feeding mechanism for the work-table. On single-spindle Besly disk grinding machines a grinding wheel is mounted at both ends of the spindle, which makes it necessary to design the bearings in such a way that provision is made for supporting the thrust load which is applied in both directions. It is also important to work out the design in such a way that there will be no end motion

of the spindle while grinding, because this would prevent the work from being ground accurately to size and would make it impossible to produce duplicate parts. End play is taken up by adjusting collars threaded on the spindle under the flange of the driving pulley. The machine can be safely operated without any end play in the spindle. In Fig. 7 it will be seen that end thrust is supported by flanges *A* and *B* of the right-hand bearing bushing. Attention is called to the fact that this bushing is made in two parts, which are separated at *C* with sufficient clearance space, so that in the event of any longitudinal expansion of the bushing, due to a rise in temperature during operation of the bearing, such expansion is taken up in the clearance space without affecting the lateral adjustment of the spindle; also, there will be no tendency for the bushing to swell outward along the spindle and bind against the thrust collars *D* and *E*, which are located at opposite ends of the radial bearing bushing. Attention is called to the fact that the bushing is anchored in the head of the grinding machine by means of two pins *F* and *G*, which fit into holes that are slightly elongated to permit expansion or contraction of the bushing without causing any unnecessary strain. There are no holes through the caps which cover each of the bearings on this machine, so that it is impossible for grit or dirt to find its way into the bearings.

In working out this design, an interesting method has been provided for lubricating both the thrust and radial bearings. Referring first to the detail view of a part of the bushing which is shown separately, it will be seen that the liner *H* is set back in order to form an oil-groove along the radial bearing at both sides of the spindle. These oil-grooves extend to the end of the bushing and then in an outward direction almost to the periphery of the flanges on the radial bearing which engage the thrust collars. The thrust faces of both flanged ends of the right-hand bearing bushing are grooved, as shown at *I*, and carry oil to all parts of the thrust bearing. The groove is circular in form, but eccentric with the spindle, and it is made of such diam-

eter that all portions of the face of each hardened thrust collar which bears against the flange on the radial bearing pass over the oil-groove during one revolution of the spindle, which insures oil reaching all parts of the thrust bearing. Oil is also delivered to this groove from the oil-grooves formed by the bushing liners *H*, to which reference has already been made. As the oil-grooves in the thrust bearings are circular in form, an eccentric with the spindle rotary motion of the thrust collars against the oil in the grooves tends to circulate this oil and provide a copious flow of lubricant. After this type of disk grinder

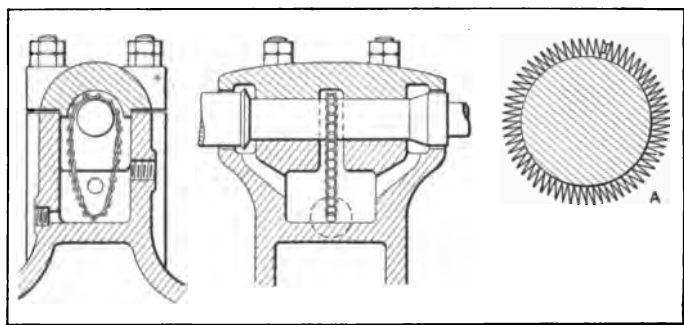


Fig. 8. Showing Lubrication by Chain and by Spring

spindle bearing was adopted, it was found practicable to increase the speed 75 per cent where such an increase was desirable.

A bearing on a grinding machine, provided with a chain-oiling device, is shown in Fig 8. The view shown at *A* represents a compromise between the chain and the solid ring—a spiral spring looped and hooked together to revolve with the shaft.

Rings for Ring-oiling Bearings. The great variety of rings that are in successful use would appear to indicate that the section of the ring that is adopted, has little to do with its efficiency. It can be seen that a ring of relatively heavy section will be less likely to be stopped by the oil than one of small section and correspondingly light

weight. Rings and oil that do acceptable work after being started often fail to start satisfactorily because the oil is stiff enough to overcome the very slight friction that exists between the shaft and the ring. This point should be considered in connection with the size of the oil reservoir and the kind of oil that gives most satisfactory results. If the reservoir is made large enough to provide sufficient oil storage to reduce the necessity of frequently renewing the oil, and the ring hangs deep in it, there will be a tendency to retard the ring when the reservoir is filled to its capacity. Large rings have a smaller area in contact with the shaft, and have a tendency to assume a position oblique to the shaft and to swing laterally; consequently the diameter of the rings should not be too large.

Chain oiling seems to offer many advantages over ring oiling, but the cheapness of rings and the fact that they give satisfactory service in millions of bearings, appears to be sufficient commendation to insure keeping them in use. Some large producers of ring oiling bearings make all their rings below 5 inches in diameter out of seamless brass tubing. The only objection to this material is that a tube is occasionally found that is eccentric enough to prevent satisfactory action. If the ring is very slightly out of balance, it will not move properly and fails to carry oil to the bearing in a satisfactory manner.

Fixed Oiling Rings or Disks. With the ring-oiling method previously referred to, the ring is hung loosely on the spindle, and revolves at a slow rate. If it is fixed to run at the same speed, the action is not sufficiently effective, because the centrifugal force throws the oil outward and very little of it can reach the shaft. When some manner of catching or scraping the oil off the ring is included, this arrangement is not objectionable. For instance, with one arrangement, the fixed ring or elevator disk throws the oil up into the distributor at the top, whence it flows along sloping surfaces which communicate by vertical holes with the top of the bearing. Some high-speed spindles have a filtering arrangement incorporated; the oil is thrown by the

disks into a trough at the top, and it runs along this into recesses filled with filtering material through which no foreign particles can pass into the bearings. Instead of providing a gage-glass for inspection, the simpler plan of screwing a brass window plate with a small piece of glass inserted is adopted. The oil-ring scraping method is represented by Fig. 9, the ring being keyed to the shaft; at the top are two lugs or scrapers, A, with only sufficient space between them for the ring to revolve. The result is that the oil raised is scraped off the ring and thrown into the grooves in the bearing.

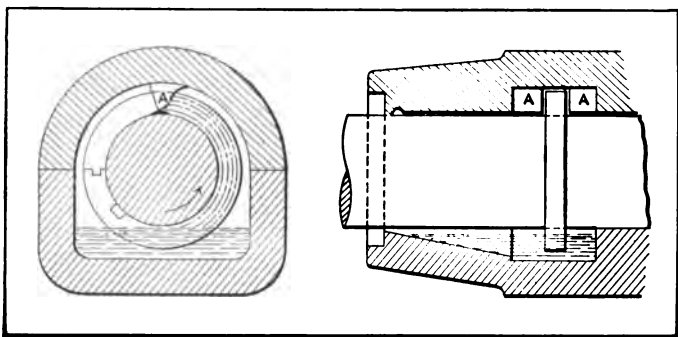


Fig. 9. Ring and Scraper Method of Lubrication which is sometimes employed with Satisfactory Results

Oil Conducted by Wicks. Wicking is now largely utilized to supply bearings in a manner that could not be accomplished by pads, or at least not so well. The principle is to trail the piece of wick in the oil reservoir, and lead it from there to the bearing surface; this is a very elastic principle, and possesses two main advantages. One is that the oil is filtered and, consequently, no dirt is transmitted by the wick; the other, that a feed can be procured from a well or gear-box not necessarily situated close to the bearing. If the wick is in proper condition, and the oil supply is maintained, there is no risk of the bearings running dry. No direct stream of oil ever reaches the bearings, since the wicking is arranged in such a fashion that the oil only climbs by capillary attraction. Fig. 10 represents the application of

long wicking to feed two bearings. A subsidiary advantage of wicking is that it can be employed to convey oil through a hole in the side of a reservoir or cup divided by a partition into two chambers. One of these is never opened to the air, and cannot receive flying grit or dust from the shop, but only forms a passage to lead the wick to the bearing surfaces. The other chamber contains the wicking coiled

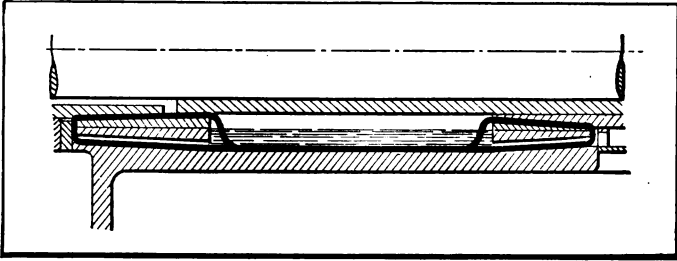


Fig. 10. Reservoir and Long Wick serving Two Bearings

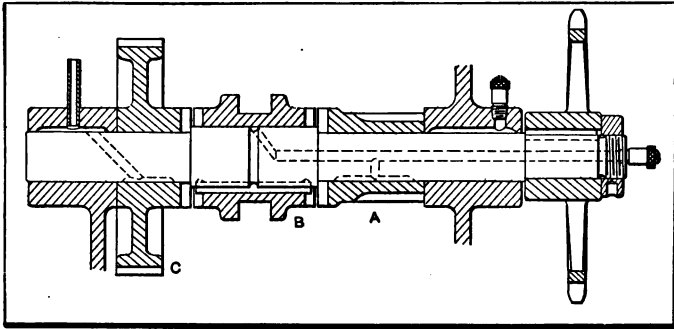
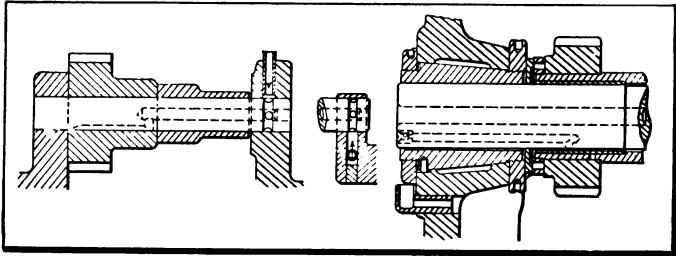


Fig. 11. Lubrication from Inside of Shaft—a Method which can be used to Advantage in Many Instances

up in a mass to absorb oil, and the communication between the chambers is such that no direct flow can occur to wash undesirable substances along.

Special Oil Ducts. There are numerous instances in which the difficulty of reaching concealed bearings results in the adoption of special pipings or passages. The advent of all-gear drives for speeds and feeds greatly complicated the matter. A common method is to drill a longitudinal hole in the shaft, and connect radial holes with this to lead out

to the various bearings or wheels. Usually the lubricant is supplied through the shaft end, but it may have to be fed from a radial or inclined hole in some cases. Fig. 11 is selected to illustrate both ways, the pinion and clutch *A* and *B* receiving oil through the shaft from the end oiler, and the gear *C* its supply from the inclined passage fed from the piped bearing adjacent. The reason for turning the semicircular groove in the shaft at the place where *B* is situated is that, as the latter does not revolve, there would be no opportunity for the oil to spread itself circumferentially. Fig. 12 shows a method of supply to a fixed pin on which is a pinion and a distance piece. A groove is



Figs. 12 and 13. Special Methods of Supplying Oil

turned in the pin where it rests in the bearing bracket, and a transverse hole leads the oil from this to the central passage. In places where it is not feasible to lead a pipe in from above, as shown, on account of the proximity of gears or other details, a lateral pipe can be inserted (see detail view, Fig. 12) to fill the vertical hole up which the oil rises to the pin. This pipe is either disposed horizontally, or slopes down, or, if a head is available from a well or tank or pump supply, it can be brought up from below. In a few instances it is impracticable to conduct oil through the center of a spindle, on account of this being used for the passage of a draw-rod or a chuck tube. In such a case, a special oil-hole is drilled, Fig. 13, in the metal between the central hole and the outside.

Piping for Conducting Lubricants. The combination of holes and pipes is frequently necessary; sometimes the oil

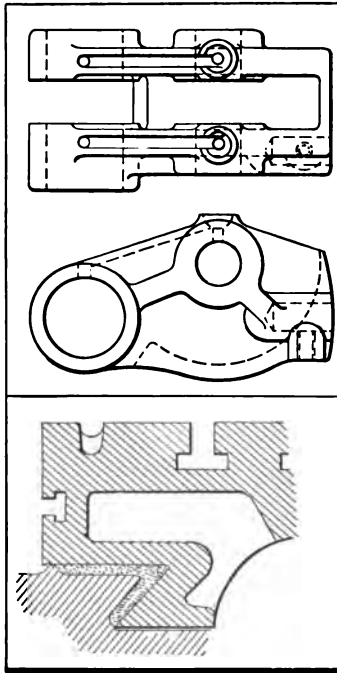
passes for a certain distance through a hole and then finishes its journey by way of a pipe for precision of location. The ease with which pipes can be bent and carried around angles greatly assists in their disposition, and often saves awkward or expensive drilling. Piping is largely utilized to span gaps where the lubricant could not be supplied with certainty, and without waste, to the interior oil-hole.

The close proximity of a pulley or other detail often renders it impracticable to reach a bearing with an ordinary oil-can spout, so the plan of arranging a pipe is followed, the top being closed with a plug, or an oiler. When a number of such pipes have to be used, it is usually best, if circumstances allow, to bring the terminations all together at one spot, not necessarily into a tank, but alongside into a holding block, and thus have them handy for attention.

The practice of covering-in sets of gears with casings which are necessary for protective purposes, or may form an integral part of the design of bearings and lever anchorages, frequently renders some amount of piping necessary, to lead from the external oilers to the various bearings. The alternative is to drill the shafts and convey the oil by way of these, but it is sometimes inconvenient to do so. The oil-pipe either leads from the cover to the bearing, or hangs some little way off and lets the oil drop into a cupped hole, or a raised rim, as the case may be. Sometimes a bushing is screwed into the bearings and has a flanged head, with an ample bell-mouth, to catch the oil. The cover can be removed and replaced without disturbing any connections, which is not the case when the pipes actually enter the bearings and fit into them.

In a great many feed-gears and other details, it is impossible to apply oil directly to certain of the bearings, because of their inaccessibility, and, in such cases, it is often the practice to provide a single trough on the top of a fixed or a tumbler casting, and drill holes and fit and bend pipes suitably to reach the other bearings. In order to prevent waste of oil, the trough is sometimes divided by partitions, thus providing each leading-out hole with its own

receptacle. When a tumbler bearing has to be lubricated, the feeding-in should be so arranged as to avoid loss of oil through the tumbler moving out of the range of the hole, or pipe, or tray from which it drops. This is done by casting a narrow trough of appropriate length on the tumbler. Fig. 14 illustrates a common design, with a cup on the higher bearing communicating by a groove with the hole



Figs. 14 and 15. Tumbler Bracket Troughs and Felt Pad for Slide

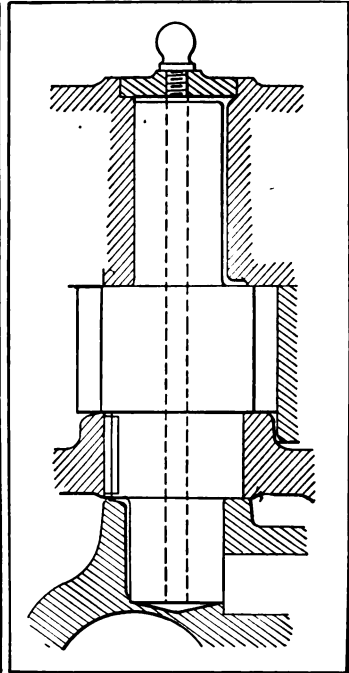


Fig. 16. Ordinary Method of Lubrication of Top and Bottom Vertical Journals

in the end hole. The groove comes under the end of the pipe through which the oil is fed, and never moves out of its reach.

In some of the more complicated designs of machines, special provisions are necessary to lubricate parts that are subject to frequent change of position—slides specifically. This is seen, for example, in the portion of a grinding ma-

chine, Fig. 17, where a sloping chute, A, receives the oil from the tube and oiler in the slide above, and conducts it over the lip of the bearing to the vertical shaft. Any alteration in position of the upper slide consequently makes no difference, and there is no waste, but all the oil from the inlet is caught. In slides not suitably provided in this manner, it may be essential to bring them to a definite position, put in the oil, and wait for a certain time to per-

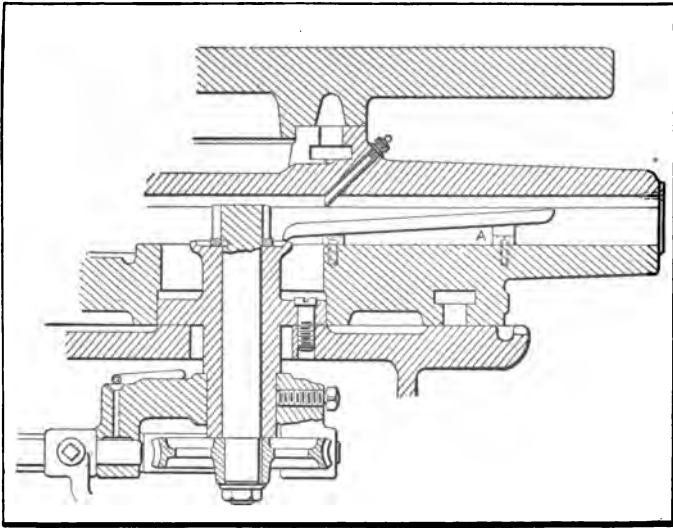


Fig. 17. Arrangement of Trough to catch Oil from Pipe Inserted in a Moving Slide above the Part to be lubricated

mit it all to escape. In this illustration the bearing for the shaft by the worm-wheel is lubricated from a bent pipe taken to the outside of the frame. Another illustration of an awkward situation is seen in Fig. 18, the vertical shaft obtaining its supply from the pipe above, discharging into the groove and thence to the bush groove, while the worm is lubricated from a combination of holes and pipes, with a trough to maintain the worm in oil constantly. Fig. 18, at A, shows a point in connection with the conduction of oil through slides, from an upper one to another; to prevent the oil from spreading in a film under the slides, the lower face

is counterbored to catch the drip from the edges of the hole and lead it properly into the continuation hole.

Lubrication of Flat Sliding Surfaces. The lubrication of tables presents some variations in methods of supply and distribution. A great deal depends upon the size and weight to be dealt with, and upon the speed of movement. A slow moving table or slide, or one subject only to occa-

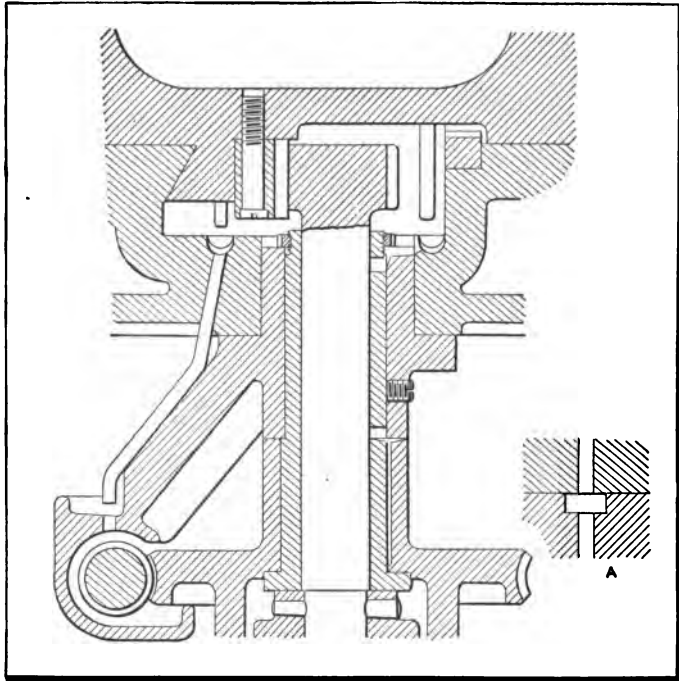


Fig. 18. Conduction by Tubes, Grooves and Passages

sional alterations in position does not require so much lubricant as a rapidly moving one in which the oil is swept off more quickly, or squeezed out by pressure. A small table, such as on a cutter grinder, for instance, can be oiled satisfactorily enough by the cam, pouring on the surfaces while the table is run back. If the length prevents this being done, lateral oil-holes are drilled, and vertical ones to communicate with the under side of the table face;

or, if V-slides are used, diagonal holes are drilled. Grooves or pads distribute the film evenly over the surfaces, and pads have the merit of retaining a portion of oil and keeping the surfaces moist for considerable periods. Several such pads can be inserted in pockets in the slide (see sectional view, Fig. 15), and the oil is then supplied by a passage to each pad.

The location of the oil-holes, plugs, or oilers is a matter of importance in certain types of tables. In some grinding machines there is no objection, for instance, to drilling the holes vertically from the table top, and letting in

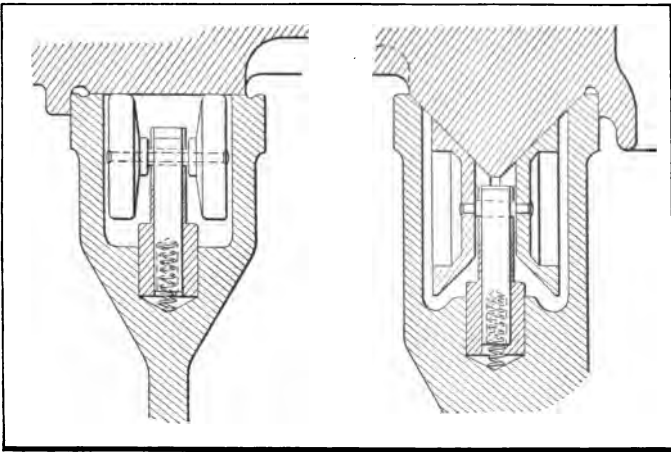


Fig. 19. Lubricating Rollers for Grinding Machine

plugs or spring caps; but in other machines these areas might be covered for a long time with fixtures or other fittings, and there would be no chance to get at the apertures without removing the fixtures. Then the lubrication might be neglected.

The most effective method of lubricating a heavy table or slide, such as that of a planer, a heavy shaper, grinder, or a boring mill, is to employ rollers sunk into oil-pockets, thus forming an automatic device independent of the care of the operator, and providing a supply of lubricant at each return of the table, or as fast as it is squeezed out. The

primitive device is simply to float a wooden roller in the oil-pocket, but it should have some means of attachment to prevent its rising unnecessarily high after the table has passed. Of course, in rotating tables, as in boring mills, a roller would remain in the same position; but, even then, it is well to afford a definite pressure by springs, which will result in proper rotation and an increase in the amount

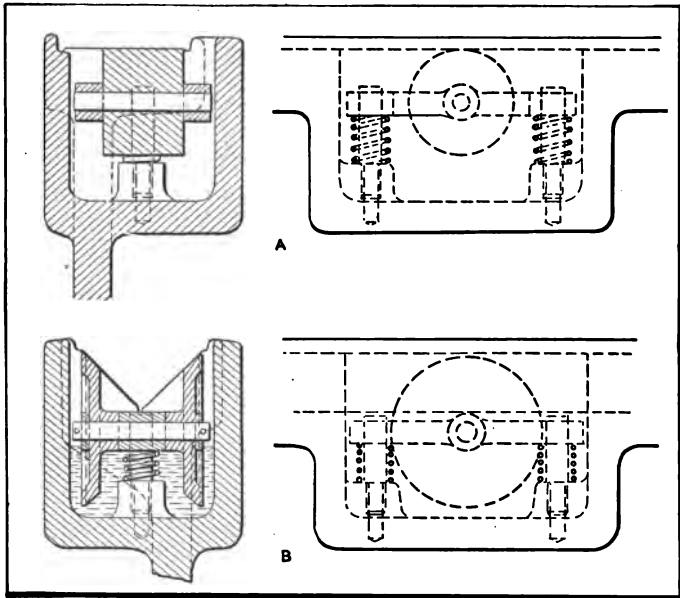


Fig. 20. Lubricating Rollers and Well for Grinding Machine, showing Application of both Plain and V-grooved Roller

of oil smeared on. Fig. 19 illustrates the type of roller fitting used in a type of grinding machines, for the flat and the V-ways, respectively. The wheels are mounted on a cross-pin held in a central plunger, which slides up and down in a casing, being maintained in the normal position at the top by the coiled spring. In another type, frames similar to those in Fig. 20, A and B, are used to lubricate the tables of grinding machines. Studs are screwed into bosses at the ends of the oil-pocket, encircled by springs, and receive the ends of the frame which supports the roller.

This construction permits the plain roller (Fig. 20, A) to be made without a break across its face. Although the V-wheels do not reach right across the slope of the vee, this is a matter of no consequence, since gravity makes up for the deficiency in this respect.

Slides which are not lubricated across their bearing width by rollers or pads require grooves to distribute the oil properly. The aim in the disposition of these grooves is to spread the oil nearly across the width, and numerous methods are followed, although the results are much the same. The precise arrangement may often depend upon the position and number of the oil-holes. In a vertical knee or

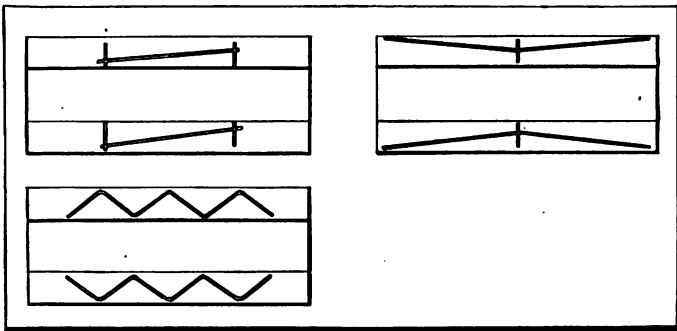


Fig. 21. Various Arrangements of Oil Grooves on Flat Surfaces

slide, with a single oiler at the center of the top edge, the grooves radiate from this oiler to the right and left. Fig. 21 gives three alternative dispositions for milling-machine tables moving horizontally. The zig-zag style is a favorite one, and is used also largely on the bearing surfaces of circular tables like those in boring mills, carrying a sufficient supply of lubricant.

Lubricating Vertical Spindles. Vertical spindles present some difficulties in regard to efficient lubrication which do not exist in horizontal ones; this is due to the inevitable tendency of the oil to run down and out of the bearings quickly. Ring-oiling is out of the question, and, if a considerable quantity of oil is required, pads must be used, or the oil must be kept in constant motion and supplied by helical

raising grooves. An ordinary method of supply is adopted in the vertical shaft illustrated by Fig. 16; the oil is poured in, on the removal of the stopper, both through the central hole and around the top of the shaft, thence flowing by the bearing grooves around the bottom and top journals. Wicks are employed extensively to conduct and distribute oil to vertical bearings. If a reservoir of oil is close at hand, the wick can be brought direct from this, and piping is not required. Felt pads are also used to retain the oil.

Lubricating Vertical Grinding Wheel Spindle Bearing. In Fig. 22 there is shown a cross-sectional view of the wheel-head of a Blanchard belt-driven vertical surface grinding machine; and the design of the bearings for carrying the wheel spindle, together with the means provided for assuring efficient lubrication, will undoubtedly prove of interest. This entire wheel-head is carried on a vertical slide, so that alignment of the spindle bearings is made entirely independent of the slide. There are ball thrust bearings at both the lower and upper ends of this spindle; and for carrying the radial load there is a bronze bearing at the lower end of the spindle and a radial ball bearing at the upper end. The great advantage of ball bearings in such a case is that they may be packed with grease and housed in such a way that the grease is kept in the bearing and adequate protection is afforded against the entrance of grit and other foreign matter, which would abrade the bearings. With such an arrangement, the bearings require practically no attention, provided they are furnished with the proper kind of lubricant which is chemically neutral; that is to say, free from both acid and alkali. Attention is called to the hemispherical seats on the races of the ball thrust bearings which fit corresponding surfaces machined to receive them. It is the purpose of this arrangement to have the hemispherical surface on the race adjust itself for slight inaccuracies in alignment or to compensate for slight strains which develop in the wheel-head, thus enabling the bearing to distribute the thrust load over all of the balls and assuring efficient transmission of power.

At the lower end of the spindle there is a tapered bearing which is arranged with the large end of the taper at the top, so that it is impossible to transmit any of the wheel thrust to this bearing. Tapered bronze bushing *A* can be easily

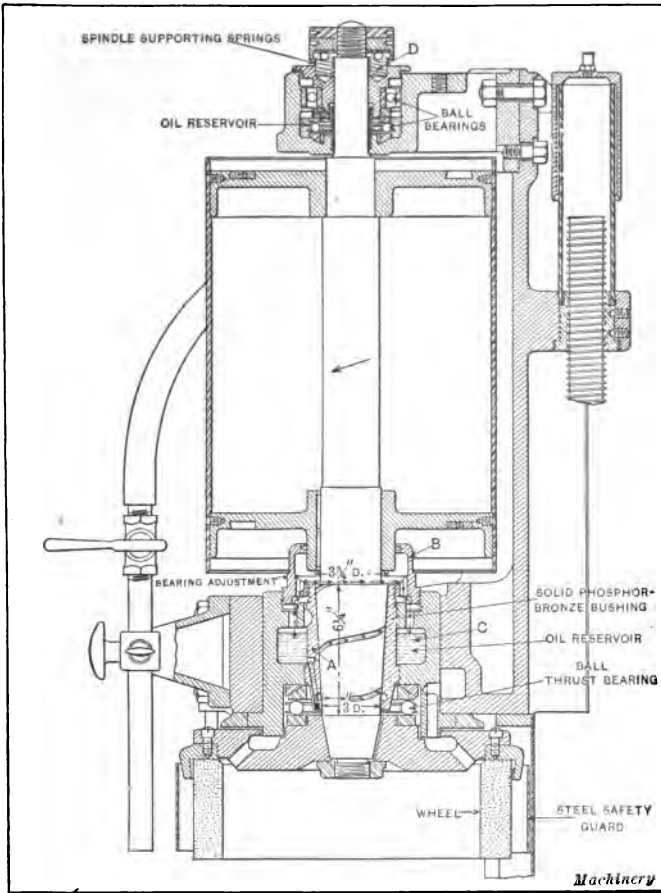


Fig. 22. Provision made for Lubrication of Spindle Bearings of Vertical Surface Grinding Machine

raised by turning the threaded ring *B* to provide for taking up any wear which develops between the spindle bearing and its bronze box. This adjusting ring *B* is turned by a spanner wrench and the bronze bushing is not split. The provision of means for delivering lubricant to this tapered

spindle bearing is particularly interesting. Lubricant is delivered to an oil reservoir *C*, from which it flows down through a channel cut in the bronze bushing at the left-hand side of the spindle to gain access to the spiral oil-groove which is cut in the journal. Oil is carried up through this groove, and in this way a liberal supply of lubricant is always distributed over the bearing. Upon reaching the top of the bearing, excess oil finds its way into channels at each side of the spindle and flows down through these channels into reservoir *C*, where it is available for subsequent use. By

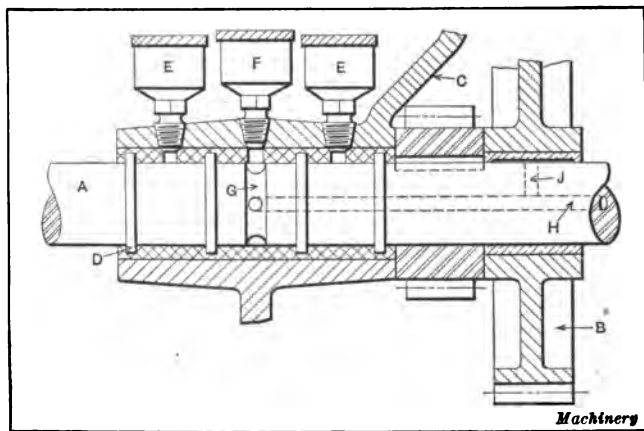


Fig 23. Lubricating Arrangement for a Gear within a Housing

studying the direction of flow, which is indicated by arrows shown in this illustration, the reader will easily obtain a perfectly clear idea of the way in which oil circulates through this bearing.

The upper end of the spindle carries a threaded collar with springs *D* beneath, which press up against it with a force exceeding the weight of the revolving parts by at least five hundred pounds. By this means, the thrust bearing at the lower end, on which depends the accuracy of the grinding, is always kept tight. All backlash in the spindle is eliminated and variations of spindle length with temperature are automatically corrected. The downward reaction

of the springs is carried on a ball thrust bearing in the upper box, and side pull at that point, due to the belt, is taken on a radial ball bearing.

Provision for Grease Lubrication. Fig. 23 illustrates provision for grease lubrication of a shaft which is journaled in a fixed frame and which has a gear *B* independently revolvable on shaft *A* (in a reverse direction to that of the shaft) and housed within a casing *C*. Rings *D* are turned on the shaft for the double purpose of taking up end thrust

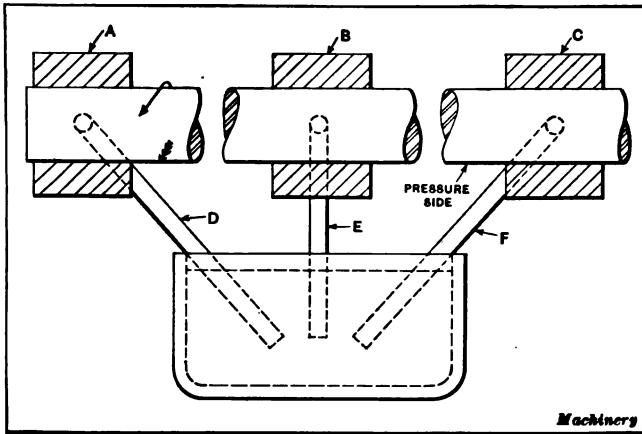


Fig. 24. Example of Multiple Capillary Lubrication

and of confining the grease which is fed between them from the cups *E*. Grease is also fed from the cup *F* into groove *G*, whence it passes through holes provided in the groove to the conductor *H*, drilled along the center of the shaft, the grease being forced out through the outlet *J* into the bearing of the gear.

Multiple Capillary Lubrication. An example of multiple capillary lubrication for inaccessible bearings is shown in Fig. 24. The bearings *A*, *B*, and *C* are supplied with oil from a tank by means of the copper tubes, *D*, *E*, and *F*, respectively, which are packed lightly with cotton wicking. The tubes extend well into the tank as may be seen, and are securely fastened in the bearings by swaging them into holes provided for this purpose. It is necessary to swage

the tubes into the bearings in order to provide an air-tight connection. It is not essential that the wicking should touch the shaft, although it should extend very close to it. The tubes should be fastened to the bearings at an angle of 90 degrees with the pressure sides, with reference to the direction of rotation of the shaft, as shown by the arrow.

Types of Self-oiling Bearings. The so-called "self-oiling" type of bearings may work on either of two principles: Oil may be taken from a reservoir and carried up to the bearing by any suitable means, or the bearing box which

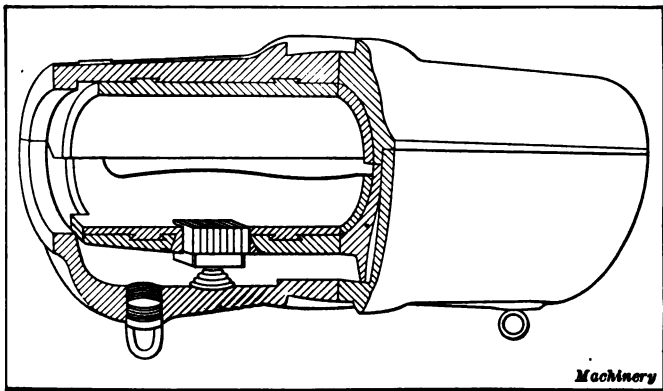


Fig. 25. Self-oiling Type of Bearings where Lubrication is effected by Capillary Action

surrounds the journal may be impregnated with some form of lubricant. The latter type of bearing is often referred to as an "oilless bearing," although in many cases it is found desirable to give bearings of this type perhaps 25 per cent of the quantity of oil which would be required for the efficient operation of a bearing of standard design. The great advantage of self-oiling or oilless bearings is that they do not require as much attention as the ordinary type, and in the event of neglect to supply such bearings with the quantity of lubricant which they are expected to receive, they are capable of operating for a considerable length of time without serious damage.

Dodge Capillary Self-Oiling Bearing. In Fig. 25 there is shown what is known as a capillary type of self-oiling bear-

ing, which is suitable for use in lineshaft hangers and similar types of equipment. This bearing is made by the Dodge Mfg. Co., of Mishawaka, Ind. The bearing box has been designed with an opening at the bottom, in which there is inserted a wooden block that extends down through the box for a sufficient distance so that the lower side of the block can dip into an oil reservoir provided in the bearing housing. The block is held against the shaft by a spiral spring, and it has a series of slots sawed in it which come to a sharp angle at one end. It is the tendency of the oil to rise in the

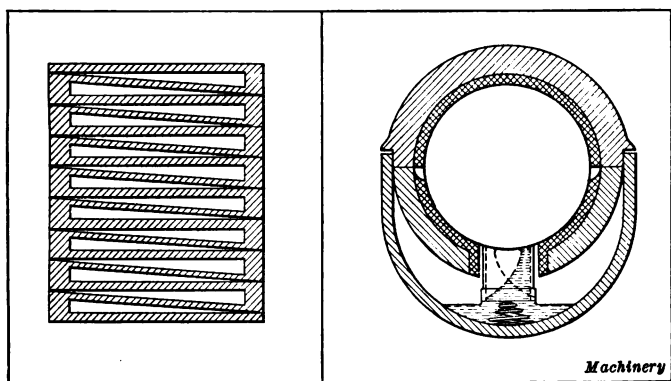


Fig. 26. Diagram illustrating Principle governing Operation of Capillary Bearing

narrow end of these slots in the wooden block, as a result of capillary action, which is responsible for carrying oil up to the bearing.

Perhaps a better idea of the arrangement will be secured after referring to Fig. 26, which shows a cross-sectional view of the wooden block, illustrating the way in which it is slotted, and a cross-sectional view through the bearing showing how the oil is drawn up in the narrow side of these slots in the wooden block, due to the capillary action which has just been mentioned. A uniform film of oil is maintained on the shaft in this way, and oil-grooves are cut in the bearing box in order that these grooves may be kept full of oil, thus facilitating lubrication of the bearing. As there is no mechanical agitation of the oil contained in the

reservoir in this bearing, any foreign matter which is present in the oil is allowed to settle to the bottom, so that only pure oil is admitted to the bearing and no trouble is experienced through grit or other foreign materials rapidly wearing the bearing surfaces. The reservoir in this bearing has sufficient capacity to contain a supply of oil which is adequate to keep the bearing efficiently lubricated for a period of six months. Before replenishing the supply of oil, the bearing should be thoroughly cleaned, and this result is easily accomplished by removing the drain plug at the bottom of the reservoir and flushing the entire bearing with kerosene, after which the plug is replaced and a fresh supply of oil placed in the bearing.

Bronze Bushing with Provision for Automatic Lubrication. Another type of self-oiling bearing is shown in Fig. 27, which is especially adapted for use in loose pulleys and similar installations where it is desirable to have the combination of a bronze bearing box and provision for automatic lubrication. This bearing is made by the Moccasin Bushing Co., of Chattanooga, Tenn. It will be seen in the illustration, which shows the method of mounting this bearing, that an oil reservoir is provided in the hub of the loose pulley, and the principle by which lubrication is effected is quite similar to that of the bearing furnished with a wick, through which oil is drawn by capillary action. Moccasin bushings can be used in any type of machine member furnished with an oil cavity from which lubricant can be drawn by the capillary oil-feeders which are contained in holes that pass transversely through the bushing so that they come into contact with the bearing surface in the manner shown on the inside of the bushing. They deliver a continuous supply of oil to the bearing, and are so prepared that they cannot glaze or clog. It is claimed that these bearings prevent the waste of oil, in addition to supplying the bearing with exactly the desired volume of oil which is required for its efficient operation.

Oil Baths for Submerging Bearing Surfaces. Submerged lubrication, that is by running parts in a bath of oil or

grease which they continually stir up and spread over themselves, exists in many forms. In the oil bath as correctly arranged, there is never any lack of lubricant and the chief care is to see that no sediment is thrown up, or that any parts are shielded from the spread of oil. The familiar worm-gear and spiral gear drive running in an oil trough was followed by the geared drives in which other classes of gears are caused to dip in oil and splash it over the teeth. Fast-running gears necessitate complete covering to prevent the oil from flying out of the box; but, in worm-gears, it is not always necessary to afford absolute protection, as, for instance, where the trough is situated within a framing, and dust or grit cannot enter. A half-trough is all that is then required. When, however, the box is situated in the open, complete enclosure is usually desirable.

The action of gears running in an oil bath can be sometimes utilized to lubricate the adjacent bearings as well, in place of fitting separate oilers to these. Care, however, must be taken that these bearings receive a sufficient amount, which is not always the case in badly designed boxes. The oil which is thrown up by centrifugal force to the roof of the box may have to be deflected by various arrangements so as to direct it into the oil recesses above the bearings. Sometimes a passage is made in the casting to lead down to these points, or ribs are cast or screwed on to catch

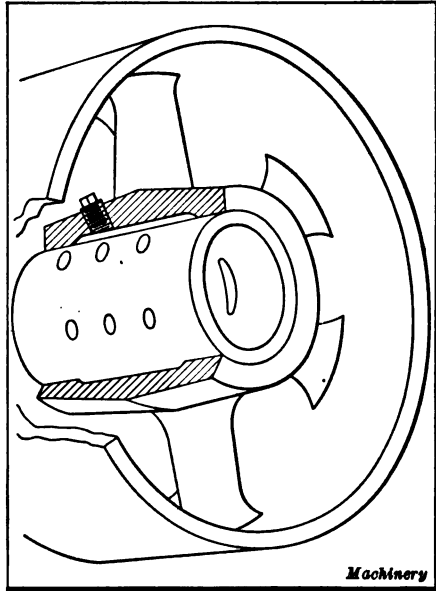


Fig. 27. Bronze Bushing provided with Means of automatically drawing Oil from Supply Reservoir

the flying oil and let it run down and drip at the place required, without actually confining it in grooves; or a piece of piping, or bent wire may be fitted and arranged to accomplish a similar function. Occasionally, a strip of metal is attached and sloped in such a manner that the oil is thrown against it, and thence off at an angle to the bearing, or to a channel communicating with it.

The ideal method of lubrication for the bearings in gear-boxes on machine tools and other bearings, where such an arrangement is possible, is to have the gears run in a reservoir filled with oil. This not only avoids excessive wear of the gear teeth, but it also provides for ample lubrication of the bearings by the splash system of lubrication. Oil-holes may be provided at the top of the bearings, so that oil thrown from the gears drops into these holes and provides a continuous flow of lubricant through the bearings. Where this arrangement is employed, an ample amount of lubricant can be put into the reservoir to last for several weeks, the length of time being governed by the conditions of service. A gage-glass on the reservoir shows the level of the oil, and at infrequent intervals it is merely necessary to replenish the supply of oil. When gears are running in mesh, or when a shaft is rotating in its bearing, the effect of friction is to tear off very fine fragments of metal which collect in the oil in such a reservoir; also, there is a tendency for a "muck" to collect in the bottom of the reservoir, due to oxidation of the oil. For these two reasons, and also owing to certain other causes, it is necessary to clean out the oil reservoirs at intervals of about six months. To facilitate this cleaning process, a drain plug should always be provided at the bottom of the reservoir which can be screwed out in order to drain off the dirty oil. The reservoir should then be flushed out with kerosene to clean away all accumulations of metal particles and oxidized oil, after which the drain plug is replaced and the reservoir refilled with clean oil.

Bearings Oiled by the Splash System. An example of the splash method of lubrication is shown in Fig. 28, which

illustrates the table gear bracket of the vertical surface grinding machine. By building this mechanism as an independent unit, the process of manufacture is simplified, in addition to making provision for the efficient lubrication of the gears and bearings by application of the splash system of oiling. The drive is transmitted through a spline shaft *A* and bevel gears to the table gear *B*. Oil is retained in reservoir *C* at the level indicated by shaded lines, and the bearing of the spline shaft is lubricated by the ring-oiling principle, which will be more fully discussed in connection with another example of bearing design. By this method,

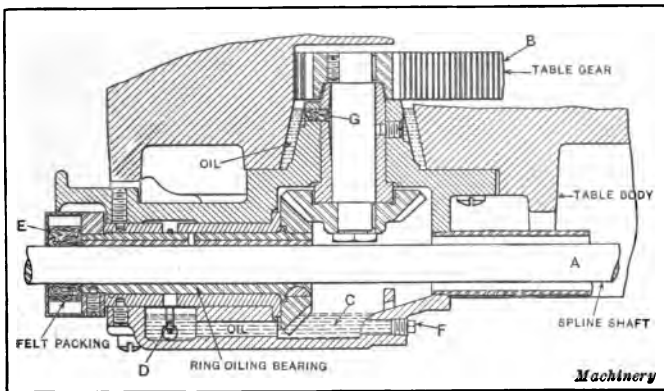


Fig 28. Method of lubricating Table Gear Bracket of Vertical Surface Grinding Machine

ring *D* runs over the bearing to be lubricated and its lower side dips into the oil carried in reservoir *C*. Rotation of the shaft results in continuously turning ring *D*, with the result that oil is carried up by the ring and deposited at the top of the bearing, thus delivering a constant supply of lubricant to the bearing. A felt packing *E* is employed to exclude abrasive dust, which would soon wear out the bearing if special precautions were not taken to prevent it from getting between the rubbing surfaces; and telescopic tubes form an effective seal at the right-hand end of the shaft. When it is required to clean out reservoir *C* by the method which has already been explained, drain plug *F* is withdrawn in order to provide for the escape of dirty

oil from the reservoir, and the flushing out of the reservoir with kerosene so as to remove all accumulated foreign matter before a fresh supply of lubricant is added.

For lubricating the bearing of the vertical shaft carrying gear *B*, oil is delivered through channels communicating with a felt packing *G* which is in contact with the bearing of the vertical shaft. This packing serves the double purpose of carrying a continuous supply of oil to the bearing and filtering out any foreign matter with which the oil may have become contaminated. Any excess oil which passes through this bearing is caught in the lower reservoir and replenishes the supply of oil for lubricating the gears and the ring-oiled bearing of the horizontal shaft. Felt packings are very generally used in this way to provide for delivering clean oil to bearings of grinding machines and similar equipments where the bearings are so situated that it is difficult to exclude abrasive dust and other substances that would rapidly score and wear out the bearings if they came in contact with the rubbing surfaces.

Flooded Lubrication. Pump systems embody many arrangements of a varied character for the thorough distribution of the lubricant. In the most complete gear-boxes, and in some machines—notably all-g geared milling machines—the same supply is utilized to flood the gears and bearings, being pumped up from the well at the base and falling from a perforated pipe in cascades on the gears, while suitably arranged pipes conduct it into the bearings. The top part of each bearing is cast as a trough, and the one below it catches the oil that runs out from the ends or center hole of the bearing. Grooves, holes, chutes, and pipes are employed to assist in the conduction.

When geared feed-boxes were first developed for use on "Milwaukee" milling machines built by the Kearney & Trecker Co., Milwaukee, Wis., it soon became apparent that in order to give efficient service, the gear-boxes would have to be provided with some means of continuous lubrication. After studying the merits of various methods, the design finally adopted consisted in constructing oil-tight

boxes, so that the gears dipped into the oil. This result proved so satisfactory that attention was turned to the idea of adopting a geared spindle drive, but to accomplish this result the gears had to be placed one above the other, thus precluding the possibility of having all of the gears dip into an oil bath. Finally, provision was made for lubrication by having the gears placed one above the other and delivering oil to the top gear by means of a pump drawing its

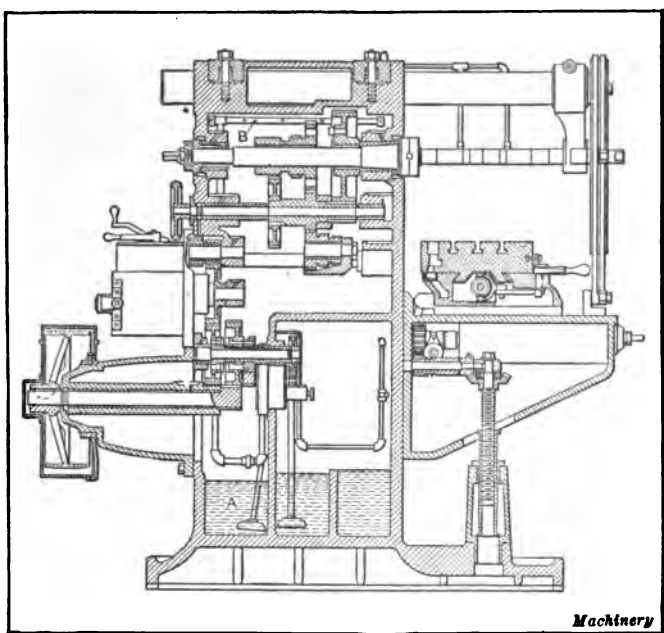


Fig. 29 Provision for delivering Continuous Stream of Oil to All Bearings and Gears in Milling Machine Drive

supply from a reservoir, as shown in Fig. 29 at A, and delivering the oil through holes in pipe B. While in operation, a gallon of oil per minute floods down over the gears and bearings. At the top of each bearing there is a cup which catches part of the oil thrown by the gears and allows it to run down through a duct to the bearings. In this way, both the gears and bearings are kept thoroughly lubricated, so that efficient transmission of power is obtained and wear

is reduced to practically the absolute minimum. In ordinary practice, the oil-grooves are closed at the ends to prevent the escape of oil from the bearings, but in the present case the grooves are cut right through to provide for a free circulation of oil and the washing away of all foreign substances from the bearings.

On 20-inch all-gearred drilling machines built by the Barnes Drill Co., Rockford, Ill., all bearings, with the exception of the spindle sleeve and cross-spindles, are continuously oiled by an automatic lubrication system. Oil is delivered by a geared pump in the reservoir of the machine and distributed constantly to all gears and bearings, including the crown gears and feed-box. This automatic lubrication system is manufactured under license from the Kearney & Trecker Co. A close view of the mechanism of one of the Barnes drilling machines is shown in Fig. 30, and the geared pump, which is driven by the constant-speed driving shaft, will be seen at the bottom of the oil reservoir. Oil is drawn into this pump through a strainer, which is also located at the bottom of the reservoir, although it is not shown in the illustration. There is a valve which can be set with just enough resistance to lift the required volume of oil to the crown gear bearings, while the remainder of the oil overflows directly into the oil-hole for the rear bearing. The pipe which will be seen just underneath the "backbone" of the machine, leading up to the crown gears, has small holes drilled in it just above each of the bearings in order that oil may escape to provide for the constant lubrication of bearings on both of the diagonal shafts. A groove is cut inside of each double bearing which leads the oil across to the lower diagonal shaft. A sufficient volume of oil is delivered to the crown gear housing so that the overflow runs into the feed-box, from which it works down through the worm-shaft bearing, and in so doing oils the worm and worm-wheel, after which it is led back into the frame of the machine and cascades back to the original reservoir. In this way, every bearing, aside from the spindle sleeve and cross-spindle bearings, is automatically supplied with a constant stream of oil.

Elevated Oil Tanks. In very large machines, when a considerable quantity of oil has to be flooded through bearings and over surfaces, a head is sometimes obtained by the use of an elevated tank, from which pipes lead down to the various grooves or passages. The pump then replenishes



Fig. 30. Provision for Continuous Lubrication of Gears and Bearings on Vertical Drilling Machine

the tank at the required rate, the oil being filtered before returning. Any blocking of the long pipes should be provided against by the inclusion of pet-cocks at suitable positions, close to the bearings, so that the position of an obstruction can be discovered. Apart from this method of obtaining a large supply, the tank is also often embodied in portions of machines to feed certain details that are

either inaccessible by an oil-can, or are too numerous for individual attention, and there is thus risk of neglect. The oil either runs direct to the places needed, or is fed slowly by siphon wicks. So long as the tank is filled, kept clean from grit or dirt, and the wicks are fitted properly, no trouble can ensue. Fig. 31 illustrates a typical case, where all the oiling orifices are located close together, and the pipes from the trough do not need to be carried far. As the trough is enclosed inside a column closed by the flap door no lid is needed to cover the oil, but this would be required if the trough were situated externally.

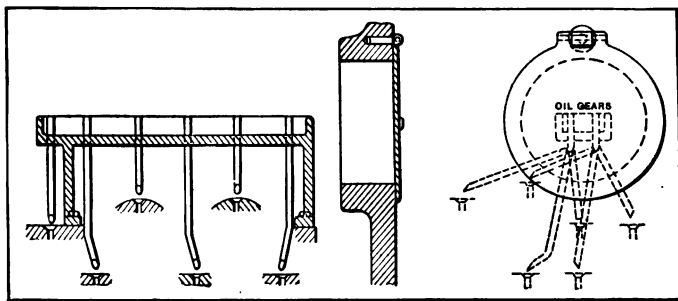


Fig. 31. Trough and Pipes supplying Several Bearings located in Proximity to each other

Oiling Lineshaft Hanger Bearings. In shops where the equipment is quite congested, it is likely to be found that there is danger of accidents to men whose duty it is to oil the bearings of linshaft hangers, countershafts, and similar equipment where the bearings are placed in somewhat inaccessible positions. A little time given to the subject of devising methods of lubrication may often be the means of saving serious accidents and consequent litigation over employer's liability. Observation of the following points will often result in saving trouble: Wherever possible, the work of oiling bearings in lineshaft hangers and countershafts should be done while the machinery is not running. A better plan is to equip such bearings with automatic lubricators which need to be filled only at infrequent intervals; with bearings of this type, it is always feasible to

arrange to replenish the supply of lubricant at some time when the machines are at rest. It will always happen, however, that certain conditions will arise, making it necessary for shafting to require attention while the machines are running, and to meet such contingencies, safe means of access must be provided, among which mention is made of the following:

Where ladders are used to reach lineshafts and countershafts, they should be provided with hooks or hangers to fit over the shaft at the upper end and spurs at the lower end, which will insure obtaining a secure grip on the floor. With such provision, there is no likelihood of a ladder slipping and causing the workman to be thrown into moving parts of a machine. In big shops it will sometimes be found feasible to have a service platform or "runway" provided with guard rails and toe-boards running parallel to lineshafts, so that any point on the shaft is readily accessible. Moving parts of machinery should not be allowed to project over such a runway, but if such a condition is unavoidable, the moving part should be completely enclosed. In some plants, use is made of a car hung from an I-beam running parallel to the lineshaft, and with such an equipment the oiler can readily move along the shaft to reach all of the bearings. The use of such a car or of the service platform is naturally restricted to quite large shops. Some shops have their men use forced-feed oil-cans of sufficient length so that they can be operated from the floor.

Where men are employed to look after the work of inspecting and oiling shafting and other equipment before the regular starting time of the plant, accidents sometimes happen through the engineer's starting up the machinery before these men have finished their work. This source of danger can be overcome by providing disconnecting appliances which make it possible to cut out any section of the equipment until the work of oiling bearings has been completed. Switches used for this purpose should be provided with padlocks, so that they can be locked in either the open or closed position. If such appliances are not in-

stalled, it is good practice to require a man employed in oiling equipment to report personally to the engineer of the plant that he is going to oil certain machinery. Probably the best safeguard in the lubrication of bearings used in all inaccessible equipment is to adopt the use of one of the so-called "self-oiling" types of bearings which can be furnished with a supply of lubricant that is adequate for a considerable period of time. Bearings of this type are furnished with means of drawing oil from the reservoir and delivering it to the bearing surface, so that constant and efficient lubrication is assured.

Oil Grooves for Bearings. In order to provide for uniform distribution of oil over the entire surface of a journal bearing, it is common practice to cut what are termed "oil-grooves" in the surface of either the journal or its box. These grooves are usually made in the form of a spiral or some kind of an endless curve, as shown in Fig. 32, which illustrates bearing boxes made by the Bunting Brass & Bronze Co., Toledo, Ohio. Special machines are made for cutting oil-grooves, which provide for handling this work in a more expeditious manner than is possible with hand tools or on machine tools of standard design. There are several reasons for cutting oil-grooves in the form of spirals or other curved shapes, instead of adopting the practice of cutting straight grooves parallel to the axis of the journal. If such a practice were followed, there would be a tendency for oil to gather in the grooves near the bottom of the box, while those on the sides and top would remain empty. With oil-grooves of spiral or similar form, however, this trouble is not experienced and rotation of the journal has the further tendency of circulating the oil through these curved grooves, thus greatly facilitating distribution of the lubricant. The oil should be introduced at that point where the forces acting tend to separate the shaft and box. At this point grooves must be cut in the surface of the box, so as to distribute the lubricant evenly over the entire length of the journal. Having been so introduced and distributed, the oil will adhere to the journal,

and be carried around by it as it revolves to the point where it is pressed against the box with the greatest force, thus forming the lubricating film which separates the rubbing surfaces. The supply of lubricant thus continually furnished, and swept up to the spot where it is needed, must not be diverted from its course in any way. A sharp edge

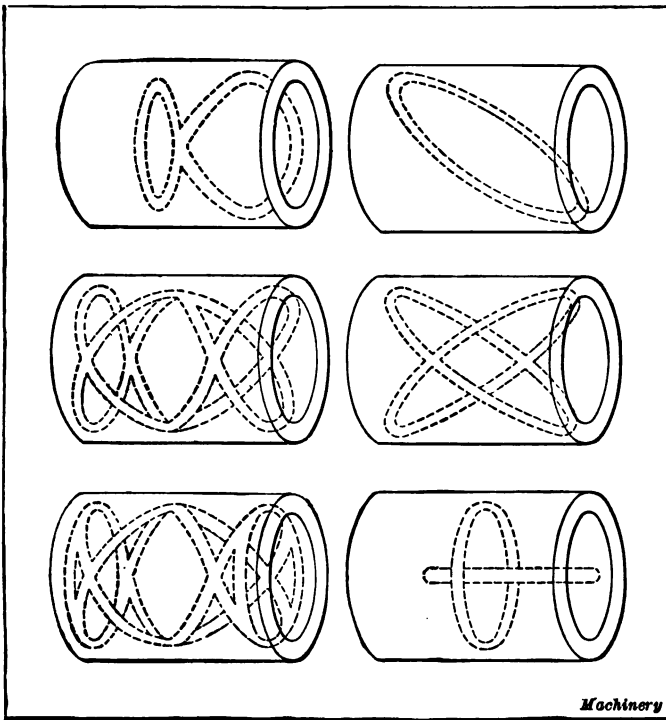


Fig. 32. Examples of Bronze Bearing Bushings showing Different Forms of Oil-grooves

at the division point of the box will wipe it off the journal as fast as it is distributed, or a wrongly placed oil groove will drain it out before it has entirely accomplished its purpose. As generally cut, oil grooves have two faults; first, they are so numerous as to cut down to a serious extent the area of the bearing, and, second, they are so located as to allow the oil to drain out of the bearing. As few grooves should be used as possible.

Two classes of bearings which may well be made without oil grooves are, first, the cross-head slippers of engines, and, second, crankpin boxes. The cross-head slipper should have a recess cut at each end, in the same way as the counterboring of the two-part box. The best way to oil a crankpin is through the pin itself. In the case of overhung pins, a hole is drilled lengthwise of the pin to its center. A second hole is drilled from the surface of the pin to meet the first one. A shallow groove should now be cut in the surface of the pin, parallel to its axis, and reaching almost to the ends of the bearing. No grooves should be cut in the boxes, but the edges where they come together should be counterbored. As much care and attention should be given to the oil grooving as to the size of a bearing, yet it is a matter often left to the fancy of the mechanic who fits it. The purpose of the grooves, to distribute the oil evenly, should ever be kept in mind, and no groove should be cut which does not accomplish this purpose, except it be to return waste oil to a place where it may again be of use. As a rule, bearings have too many grooves and, far from assisting the lubricants, they generally drain the oil from where it is most needed.

CHAPTER IV

DESIGN OF PLAIN BEARINGS

THE design of journals, pins, and bearings of all kinds is one of the most important problems connected with machine construction. It is a subject upon which there is, unfortunately, conflicting data. The results obtained from the rules given by different engineers will be found to differ by 60 per cent or more. Many of the best modern engines have been designed in defiance of the generally accepted rules on this subject, and many other engines, when provided with what were thought to be very liberal bearing surfaces, have proved unsatisfactory. This confusion has largely been the result of a misconception of the actual running conditions of bearing.

In working out the design of journal bearings for use on any type of machine, it is necessary for careful consideration to be given to the conditions of service under which each bearing will operate. This is particularly true in the case of bearings used on machine tools, because there is a great amount of variation in the conditions which machine tool bearings are required to fulfill. In bearings for driving shafts, provision made for the efficient transmission of power and freedom from wear are usually the two points of maximum importance; but in the main spindle bearings, these important points may receive consideration only after provision has been made in the design to assure obtaining a bearing which will be a tight running fit when new and in which means are provided to compensate for any wear that may develop after the machine is placed in service. Lost motion in a spindle bearing will show itself by the presence of chatter marks on the work and by difficulty experienced in holding dimensions of the work within the required limits of accuracy. On this account, the designer of machine

tool spindle bearings must first consider the conditions under which the bearings are to operate with the idea of fulfilling these requirements without tendency for the bearings to wear excessively.

After this has been done, it is necessary to take steps to work out the design along such lines that convenient means are provided for readily taking up any small amount of wear which may develop in the bearings. Only after provision has been made for maintaining an accurate fit between the spindle and its bearings can the designer turn his attention to the question of transmission efficiency, although this condition must also be fulfilled. There are often special conditions which must be considered in designing bearings in order to assure satisfactory operation. For instance, on grinding machines, trouble would almost surely be experienced through abrasive dust from the wheel finding its way into the bearings and causing them to wear excessively, unless the housings were designed with special provision to exclude dust and other foreign matter from the bearings. This is an example of special conditions likely to require careful consideration in working out the design of bearings capable of giving satisfactory service.

General Principles Covering Bearing Design. A journal should be designed of such a size and form that it will run cool, and with practically no wear. The question of both heating and wear is one of friction, and in order to understand the principles upon which the design of bearings should be based, the underlying principles of friction must first be understood. Friction is defined as that force acting between two bodies at their surface of contact, when they are pressed together, which tends to prevent their sliding one upon the other. The energy used in overcoming this force of friction appears at the rubbing surfaces as heat, and is ordinarily dissipated by conduction through the two bodies. The force of friction, and hence the amount of heat generated under any given circumstances, can be greatly reduced by the introduction of an oily or greasy substance between the rubbing surfaces. The oil or grease seems to act in the

same way that a great number of minute balls would, reducing the friction and wear, and thus preventing the overheating and consequent destruction of the parts. On this account, all bearings are always lubricated. Thus the question of journal friction involves the question of lubrication.

For the purpose of understanding as far as possible what goes on in a bearing, and the amount and nature of the forces acting under different conditions, several machines have been designed to investigate the matter. In general, they are so arranged that a journal may be rotated at any desired speed, with a known load upon the boxes. Suitable means are provided for measuring the force of friction and also the temperature of the bearing. From investigations with such apparatus, it has been found that the laws of friction of lubricated journals differ very materially from those commonly stated in text-books as the laws of friction.

Frictional Resistance in Lubricated and Unlubricated Bearings. It is generally stated that the force of friction is proportional to the force with which the rubbing surfaces are pressed together, doubling, or trebling, as the case may be, with the normal pressure. This law is perfectly true for all cases of unlubricated bearings, or for bearings lubricated with solid substances, such as graphite, soapstone, tallow, etc. When, however, the bearing is properly lubricated with any fluid, it is found that doubling the pressure does not by any means double the friction, and when the lubricant is supplied in large quantities, by means of an oil bath or a force pump, the friction will scarcely increase at all, even when the pressure is greatly increased. From the experiments of Thurston, and also of Tower, it appears that the friction of a journal per square inch of bearing surface, for any given speed, is equal to:

$$f = kp^n \quad (1)$$

where f = the force of friction acting on every square inch of bearing surface;

p = the normal pressure in pounds per square inch on that surface;

k = a constant.

The exponent n depends on the manner of oiling, and varies from 1, in the case of dry surfaces, to 0.50, in the case of drop-feed lubrication; 0.40 or thereabouts in the case of ring- and chain-oilers and pad lubrication; and becomes zero in case the oil is forced into the bearing under sufficient pressure to float the shaft.

The second law of friction, as generally stated, is that the force of friction is independent of the velocity of rubbing. This law also is true for unlubricated surfaces, and for surfaces lubricated by solids. In the case of bearings lubricated by oil, the friction increases with the speed of rubbing, but not at the same rate. Expressed as an equation:

$$f = kv^m \quad (2)$$

where f = the force of friction at the rubbing surfaces in pounds per square inch;

k = a constant;

v = the velocity of rubbing in feet per second.

The exponent m varies from zero, in the case of dry surfaces, to 0.20, in the case of drop-feed, and 0.50, in the case of an oil bath.

The third law of friction, as it is generally stated, is that the friction depends, among other things, on the composition of the surfaces rubbed together. This, again, while true for unlubricated surfaces, is not true for other conditions. No matter whether the surfaces be steel, brass, babbitt, or cast iron, so long as they are perfectly smooth and true, they will have the same friction when thoroughly lubricated. The friction will depend upon the oil used, not on the materials of journal or boxes, when the other conditions of speed and pressure remain constant. Many people think that babbitt has less friction than iron or brass, under the same circumstances, but this is not true. The reason for the great success of babbitt as an "anti-friction" metal depends upon an entirely different property, as will be explained later.

Combining into one equation the different laws of the friction of lubricated surfaces:

$$f = kp^n v^n \quad (3)$$

where f = the force of friction at the rubbing surface in pounds per square inch;

k = a constant, which varies with the excellence of the lubricant from 0.02 to 0.04.

The other quantities are as before. From this expression it is evident that the friction increases with the load on the bearing, and also with the velocity of rubbing, although much more slowly than either.

Generation of Heat in Bearings. The quantity of heat generated per square inch of bearing area, per second, is equal to the force of friction times the velocity of rubbing. All of this heat must be conducted away through the boxes as fast as it is generated, in order that the bearing shall not attain a temperature high enough to destroy the lubricating qualities of the oil. The hotter the boxes become, the more heat they will radiate in a given time. When the bearing is running under ordinary working conditions, it will warm up until the heat radiated equals the heat generated, and the temperature so attained will remain constant as long as the conditions of lubrication, load, and speed do not change. This rise in temperature above that of the surrounding air varies from less than 10 to nearly 100 degrees F., and is commonly about 30 degrees F. The force of friction or the velocity of rubbing, or both, must be kept down to that point where the temperature shall not attain dangerous values. As has been shown in the preceding paragraph, it was formerly believed that the force of friction was equal to a constant times the bearing pressure, and, therefore, that the work of friction was equal to this constant times the pressure, times the velocity of rubbing. Now, since it is the work of friction that must be limited to a certain definite value per square inch of bearing area, it has been concluded that a bearing would not reach a dangerous temperature if the product of the bearing pressure per square inch and the velocity of rubbing did not exceed a certain value. The limiting value for the product of pressure and velocity is, therefore, determined by Thurston's formula:

$$pv = C \quad (4)$$

in. which p = bearing pressure, in pounds per square inch;
 v = velocity of rubbing, in feet per second.

C = constant having values varying from 800 foot-pounds per second, in the case of iron or low-carbon steel shafts, to 2600 foot-pounds, in the case of high-carbon steel crankpins.

This formula is satisfactory for bearings running at ordinary speeds, although it must be modified for extreme cases.

The results obtained from the machines for testing bearings are very even and regular for ordinary pressures and temperatures, but when either of these is increased to a high point, the friction and wear of the bearing suddenly increases enormously. The reason is that the oil has been squeezed out of the bearing by the great pressure. This squeezing out of the oil and consequent great increase in the friction has three effects. The absence of the lubricant causes the parts to scratch or score each other, thus rapidly destroying themselves, the great increase in friction results in a sudden very high temperature, in itself destructive to the materials of the bearing, and the heating is generally so rapid as to cause the pin and the interior parts of the box to expand more rapidly than the exterior parts, thus causing the box to grip the pin with enormous pressure. When the oil has been squeezed out in this manner, the bearing is said to "seize."

Influence of Quality of Oil. The unit pressure which any bearing will stand without seizing depends upon its temperature and the kind of oils used. The lower the temperature of the bearings, the greater the allowable unit pressure. The reason for this is that oils become thinner and more free-flowing at the higher temperatures, consequently they are more easily squeezed out of the bearing, and it is more likely to seize. On this account, the higher the velocity of rubbing, the less the unit pressure that can be carried, but it does not follow that the allowable unit pressure varies inversely as the speed of rubbing, as was formerly thought. The thicker and less free-flowing an oil is, the greater the

unit pressure it will stand in a bearing without squeezing out. A watch oil or a light spindle oil can be used only under a very small unit pressure. Sometimes they are squeezed

Bearing Pressures for Various Classes of Bearings (Kimball and Barr)

Class of Bearing and Condition of Operation	Allowable Bearing Pressure, Lb. per Sq. In.
Bearings for very slow speed as in turntables in bridge work.....	7000 to 9000
Bearings for slow speed and intermittent load as in punch presses.....	3000 to 4000
Locomotive wrist-pins	3000 to 4000
Locomotive crankpins	1500 to 1700
Locomotive driving journals.....	190 to 220
Railway car axles.....	300 to 325
Marine engine main bearings. { naval practice	275 to 400
{ merchant practice	400 to 500
Marine engine crankpins.....	400 to 500
Stationary engine main bearings { for dead load*	60 to 120
(high speed) { for steam load	150 to 250
Stationary engine crankpins { overhung crank	900 to 1500
(high speed) { center crank	400 to 600
Stationary engine wrist-pins (high speed).....	1000 to 1800
Stationary engine main bearings { for dead load*	80 to 140
(slow speed) { for steam load	200 to 400
Stationary engine crankpins (slow speed).....	800 to 1300
Stationary engine wrist-pins (slow speed).....	1000 to 1500
Gas engines, main bearings.....	500 to 700
Gas engines, crankpins.....	1500 to 1800
Gas engines, wrist-pins.....	1500 to 2000
Heavy lineshaft brass or Babbitt lining.....	100 to 150
Light lineshaft cast-iron bearing surfaces.....	15 to 25
Generator and Dynamo bearings.....	30 to 80

*Weight of shaft, flywheels, etc.

out of the bearing when the pressure does not exceed 50 pounds per square inch. A cylinder oil of good body will stand a pressure of over 2000 pounds per square inch in the same bearing. It is important to determine, if possible, in each case, what quality of oil is best adapted to each par-

ticular bearing. A third cause influencing the pressure which may be carried is adhesiveness between the oil and the rubbing surfaces. Some oils are more certain to wet metal surfaces than are others, and, in the same way, some metals are more readily wet by oil than are others. It is evident that when the surfaces repel, rather than attract, the oil, the film will be readily broken down, and, when the opposite is the case, the film is easily preserved.

Calculating Bearing Dimensions. The durability of the lubricating film is affected in great measure by the character of the load that the bearing carries. When the load is unvarying in amount and direction, as in the case of a shaft carrying a heavy flywheel, the film is easily ruptured. In those cases where the pressure is variable in amount and direction, as in railway journals and crankpins, the film is much more durable. When the journal rotates through only a small arc, as with the wrist-pin of a steam engine, the circumstances are most favorable. It has been found that when all other circumstances are exactly similar, a car journal, where the force varies continually in amount and direction, will stand about twice the unit pressure that a flywheel journal will, where the load is steady in amount and direction. A crankpin, since the load completely reverses for every revolution, will stand three times, and a wrist-pin, where the load only reverses but does not make a complete revolution, will stand four times the unit pressure that the flywheel journal will.

The amount of pressure that commercial oils will endure at low speeds without breaking down varies from 500 to 1000 pounds per square inch, where the load is steady. It is not safe, however, to load a bearing to this extent, since it is only under favorable circumstances that the film will stand this pressure without rupturing. On this account, journal bearings should not be required to stand more than two-thirds of this pressure at slow speeds, and the pressure should be reduced when the speed increases. The approximate unit pressure which a bearing will endure without seizing is as follows:

$$p = \frac{PK}{DN + K} \quad (5)$$

in which p = allowable pressure in pounds per square inch of projected area;

D = diameter of bearing in inches;

N = number of revolutions of journal per minute;

P = maximum safe unite pressure under given circumstances at slow speed. Ordinarily the value of P is 200 for collar thrust bearings; 400 for shaft bearings; 800 for car journals; 1200 for crankpins; and 1600 for wrist-pins. Under exceptional circumstances, these values may be increased by as much as 50 per cent, but only when the the workmanship is the best and the care of the bearing the most skillful; in addition, a bearing should be readily accessible and the oil of the best quality and unusually viscous. Only in the case of very large machinery having expert supervision, can these higher values be safely adopted;

K = quantity depending on method of oiling, etc.

Its value may be assumed for ordinary work, drop-feed lubrication, as 700; first-class care, drop-feed lubrication, 1000; for force-feed lubrication or ring oiling, from 1200 to 1500; extreme limit for perfect lubrication and air-cooled bearings, 2000. The value of 2000 is seldom used except in locomotive work where the rapid circulation of the air cools the journals. Higher values than 2000 may be used only in the case of water-cooled bearings.

Formula (5) is in a convenient form for calculating journals. In case the bearing is some form of sliding shoe, the quantity $240 V$ should be substituted for the quantity DN , V being the velocity of rubbing in feet per second.

There are a few cases where a unit pressure sufficient to break down the oil film is allowable. Such cases are the pins of punching and shearing machines, pivots of swing bridges and similar constructions, where the motion is slow and heating cannot well result. In such cases, pressures up to 4000 pounds per square inch are permissible.

High-speed Bearings. In carefully lubricated high-speed bearings, very high unit pressures are permissible. The following figures represent the practice of a large concern building electrical machinery in large units:

Velocity in feet per second of

rubbing surfaces	20	30	40	60	75
------------------------	----	----	----	----	----

Permissible pressure in pounds

per square inch	165	190	205	225	230
-----------------------	-----	-----	-----	-----	-----

The permissible pressures here increase with the speed. The reason for this is that the higher the surface speed, the more effectively is the lubricant dragged in against the hydraulic pressure due to the load carried, the high-speed bearing acting in a measure as a pump for its lubricant.

Diameter of Shaft or Pin. The diameter of a shaft or pin must be such that it will be strong and stiff enough to carry the load. In order to design it for strength and stiffness, the approximate length must be known. This will be assumed from the following equation:

$$L = \frac{20 W \sqrt{N}}{PK} \quad (6)$$

in which L = length of bearing in inches;

W = total load upon bearing in pounds;

N = number of revolutions of journal per minute;

P and K = same quantities as in Equation (5).

When the approximate length has been found by the use of this equation, the diameter of the shaft or pin may be found by the general formulas for the strength of materials. The length of the journal must then be recomputed by the formula given in the next paragraph.

Length of Bearing. Having obtained the proper diameter, a bearing length must be selected long enough so

that the unit pressure shall not exceed the required value. This length may be found directly from the equation:

$$L = \frac{W}{PK} \left(N + \frac{K}{D} \right) \quad (7)$$

in which L = length of bearing in inches;

W = total load upon bearing in pounds;

P , K , N , and D = same quantities as in Equation (5).

Should the length obtained by this formula not give practical dimensions for proper proportions, the diameter and length of the bearing must be adjusted to meet the conditions. A good rule for the length of the journal, after the

Relation of Length l to Diameter d of Journals

Type of Bearing	Values of $\frac{l}{d}$
Marine engine main bearings.....	1 to 1.5
Marine engine crankpins.....	1 to 1.5
Stationary engine main journals.....	1½ to 2.5
Stationary engine crankpins.....	1
Stationary engine cross-head pins.....	1 to 1.5
Ordinary heavy shafting with fixed bearings.....	2 to 3
Ordinary shafting with self-adjusting bearings.....	3 to 4
Generator bearings	3

requirements with relation to the bearing pressures have been met, is to make the ratio of the length to the diameter about equal to $\frac{1}{8}$ of the square root of the number of revolutions per minute. This quantity may be decreased from 10 to 20 per cent in the case of crankpins and increased in the same proportion in the case of shaft bearings, but should not be departed from too widely. In the case of an engine making 100 revolutions per minute, the length of the bearings would, by this rule, be from one and one-quarter to one and one-half times the diameter. In the case of a motor running at 1000 revolutions per minute, the bearings would be about four diameters long. This rule, while it cannot be adhered to on all occasions, is an excellent guide.

A bearing may give poor satisfaction because it is too long, as well as because it is too short. Almost every bearing is in the condition of a loaded beam, and, therefore, has some deflection. Take the case of an overhung crankpin, in order to examine the phenomena occurring in a bearing under these circumstances. When the engine is first run, both the pin and box are, or should be, truly round and cylindrical. As the pin deflects under the action of the load, the pressure becomes greater on the side toward the crank throw, breaking down the oil film at that point, and causing heat. After a while the box, therefore, becomes worn to a slightly larger diameter at the side toward the crank. The box, as already mentioned, must be a trifle larger in diameter than the journal, and, for successful working, this difference is very strictly defined, and can vary only within narrow limits. Should the pin be too large, the oil film will be too thin, and easily ruptured. On the other hand, should the pin be too small the bearing surface becomes concentrated at a line, and the greater unit pressure at that point ruptures the film. This is also the case when the pin is too long. The box rapidly wears large at the inner end, and, consequently, the pressure becomes concentrated along a line. The lubricating film then breaks down, and the pin heats and scores. The remedy is not to make the pin longer, so as to reduce the unit pressure, but to decrease its length and to increase its diameter, causing the pressure to be evenly distributed over the entire bearing surface.

Examples of Calculating Dimensions for Bearings. A few examples will serve to make plain the methods of designing bearings by means of these principles.

Examples: Design a collar thrust bearing for a 10-inch propeller shaft running at 150 revolutions per minute, and with a thrust of 60,000 pounds. Assuming that the thrust rings will be 2 inches wide, their mean diameter will be 12 inches. From Equation (5) the allowable bearing pressure is:

$$\frac{200 \times 700}{12 \times 150 + 700} = 56 \text{ pounds per square inch,}$$

requiring a bearing of $60,000 \div 56$, or 1070 square inches area. Since each ring has an area of 0.7854 ($14' - 10'$), or about 75 square inches, the number of rings needed will be $1070 \div 75$, or 14. In case it were desirable to keep down the size of this bearing, the constant K might have been given values as high as 1000 instead of 700.

Example: Design a main bearing for a horizontal engine. Assume that the diameter of the shaft is 15 inches and that the weight of the shaft, flywheel, crankpin, one-half of the connecting-rod, and any other moving parts that may be supported by the bearings, is 120,000 pounds, and that two-thirds of this weight comes on the main bearing, the remainder coming on the outboard bearing. The engine runs at 100 revolutions per minute. In this case, $W = 80,000$ pounds, $P = 400$ pounds per square inch, and K depends upon the care and method of lubrication. Assuming that the bearing will be flushed with oil by some gravity system, and that, since the engine is large, the care will be excellent, take $K = 1500$. This gives the length of the bearing from Formula (7) :

$$L = \frac{80,000}{400 \times 1500} \left(100 + \frac{1500}{15} \right) = 26\frac{1}{2} \text{ inches, approximately.}$$

In computing the length of this bearing, the pressure of the steam is not considered since it is not a steady pressure; but the projected area of the main bearing must be greater than the projected area of the crankpin.

Example: Find the dimensions for the bearings of a 100,000-pound hopper car weighing 40,000 pounds, having eight 33-inch wheels. The journals are $5\frac{1}{2}$ inches in diameter, and the car is to run at 30 miles per hour. The wheels will make 307 revolutions per minute when running at this speed, and the load on each journal will be $140,000 \div 8$, or 17,500 pounds. Although the journal will be well lubricated by means of an oil pad, it will receive but indifferent care, so the value of K will be taken as 1200. Using these values, the length of the journal will then be determined as follows:

$$L = \frac{17,500}{800 \times 1200} \left(307 + \frac{1200}{5.5} \right) = 9\frac{5}{8} \text{ inches, approximately.}$$

Example: Design a crankpin for an engine with a 20-inch steam cylinder running at 80 revolutions per minute, and having a maximum unbalanced steam pressure of 100 pounds per square inch. The maximum and not the mean steam pressure should be taken in the case of crank- and wrist-pins. The total steam load on the piston is 31,400 pounds. P will be taken as 1200, and K as 1000. By Formula (6):

$$L = \frac{20 \times 31,400 \times \sqrt{80}}{1200 \times 1000} = 4.7, \text{ or say, } 4\frac{3}{4} \text{ inches.}$$

In order that the deflection of the pin shall not be sufficient to destroy the lubricating film:

$$D = 0.09 \sqrt[3]{WL^3}$$

which limits the deflection to 0.003 inch. Substituting in this equation, the diameter becomes 3.85, or, say, $3\frac{7}{8}$ inches. With this diameter, obtain the length of the bearing, by using Formula (7):

$$L = \frac{31,400}{1200 \times 1000} \left(80 + \frac{1000}{3\frac{7}{8}} \right) = 8.85, \text{ or say, } 9 \text{ inches.}$$

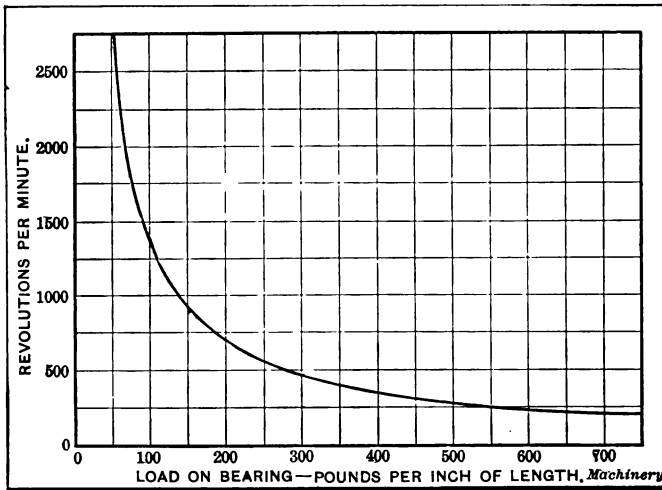
The mean of this value and the one obtained before is about 7 inches. Substituting this in the equation for the diameter, $D = 5\frac{1}{4}$ inches. Substituting this new diameter in Equation (7):

$$L = \frac{31,400}{1200 \times 1000} \left(80 + \frac{1000}{5\frac{1}{4}} \right) = 7.1, \text{ or say, } 7 \text{ inches.}$$

It would now be preferable to take about half an inch off the length of this pin, and add it to the diameter, making it $5\frac{3}{4} \times 6\frac{1}{2}$ inches, and this will be found to bring the ratio of the length to the diameter nearer to one-eighth of the square root of the number of revolutions.

Chart for Safe Load on Journal Bearings. In overcoming the friction between a journal and its bearing, a certain

amount of energy is expended in the form of heat, as previously explained. The only means of getting rid of this heat is by conduction from the heat generating surfaces, through the masses of the shaft, bearing and bearing support, and thence by radiation to the outside air. The oil to a certain extent serves as a medium for the transfer of a part of the heat generated, by giving up its heat to the walls of the oil chamber; but by far the greater part is dissipated by conduction in the manner described.



Curve giving Safe Load per Inch of Bearing Length

Inasmuch as there is always some heat generated in the bearing, no matter how liberally it may be designed, there must always be a temperature increase in the bearing. If this heat can be carried off as fast as it is generated, the bearing will at some time in its operation reach a constant temperature. As the radiating capacity of any bearing is a fixed quantity, the temperature reached will depend on the rate at which heat is generated; therefore, in order that a bearing may not overheat, there must be some means of determining this rate, and fixing a limit to which the amount of heat generated may be carried. Between bearing surfaces, as in all cases where relative motion takes place be-

tween two surfaces in contact, the work done in overcoming friction depends on the rate of motion and the pressure existing between the two surfaces. This is modified to a certain extent by the character of the surfaces in contact, but for practical purposes, it is sufficient to consider the two factors of speed and pressure, and as the work done is proportional to each it is proportional to their product. By fixing the product of these factors as a constant, we have a means of limiting the work done in the bearing to a safe quantity.

It has been found by observation of a large number of bearings that were known to operate within a temperature rise of 40 degrees C. that if the rotative speed is taken in feet per minute, and the pressure in pounds per square inch of projected bearing area, the constant lies between the limits of 36,000 and 40,000. If we use the smaller number and let

d = diameter of bearing in inches,

l = length of bearing in inches,

L = total load on bearing in pounds,

$\frac{L}{l} = P$ = load on bearing per inch of length,

N = revolutions per minute,

we have,

$$\frac{\pi d N}{12} \times \frac{L}{dl} = 36,000$$

which reduces to

$$\frac{NL}{l} = 137,000, \text{ or } NP = 137,000$$

From this we see that if the product of the revolutions per minute and the load on the bearing per inch of length does not exceed 137,000, the bearing will operate within the temperature rise specified. By plotting these factors, the accompanying curve was obtained, which presents this formula in a convenient shape for use. By reading across to the curve

from revolutions per minute at the left, and thence down, the safe load per inch of bearing length will be found.

In determining the pressure on the bearing, any load due to belt or chain pull, or gear thrust, in addition to the direct weight of the shaft and member carried by it, must be taken into account. In the majority of cases the belt pull will be at an angle to the direct weight on the bearing; the gear thrust may be in any direction—most frequently, however, in the same direction or opposite to the direct weight. In any case, a resultant of the forces operating on the bearing must be taken and used as the final load figure.

Allowance for Oil Between Shaft and Journal in Medium and High-Speed Bearings

Diameter of Journal	Allowance	Diameter of Journal	Allowance	Diameter of Journal	Allowance
$\frac{3}{8}$ to 1	0.002	$3\frac{5}{8}$ to $4\frac{1}{2}$	0.005	6	0.009
$1\frac{1}{8}$ to $2\frac{1}{2}$	0.003	5	0.006	7	0.011
$2\frac{5}{8}$ to $3\frac{1}{2}$	0.004	$5\frac{1}{2}$	0.007	8	0.012

It will be noted that the diameter of the bearing does not have to be considered in determining its safe load. While it enters into the formula as a factor of the rotative speed, it also has the function of directly reducing the pressure per square inch of bearing area, so that it is eliminated from the formula in its final shape. The diameter of the bearing is generally fixed from other considerations, being chiefly dependent upon the size of shaft required for stiffness, and for transmitting the given horsepower.

Clearance between Journals and Bearing Boxes. In designing journals and the bearing boxes in which they are to run, care must be taken to proportion the sizes of the journal and its box in such a way that just a sufficient amount of clearance will be left to provide the necessary space for a film of oil which is required in the bearing. This amount of clearance varies with the diameter of the shaft, and in the accompanying table is given the clearance which should be left for shafts of different sizes:

The allowances of some manufacturers are much smaller than those given in the table, as indicated by the following formula: The allowance made for the "running fit" of the box and shaft should be about $0.0005 (D + 1)$ inch, where D is the nominal diameter of the shaft in inches.

Frictional Losses in Babbitt, Ball, and Roller Bearings. In order to determine the relative power consumption for babbitt, ball and roller bearings, a series of tests was made and the results given in a paper by Carl C. Thomas, E. R. Maurer and L. E. A. Kelso, presented before the American Society of Mechanical Engineers.

The object of these tests was to ascertain definitely the relative and absolute amounts of power required to drive a specially constructed lineshaft carrying given loads at certain known speeds of revolution, when supported successively by the three different types of shaft bearings mentioned, and to determine coefficients of friction for each type. Twenty bearings of each type were used in order that representative results might be obtained.

The design of the apparatus was made with the assistance of the manufacturers of the bearings to whom preliminary drawings were submitted, and during the four years of the tests representatives of these firms visited the laboratory for the purpose of giving whatever advice and assistance was possible. The preliminary work, covering the first two years, showed the necessity of considering the temperature of the oil film in the babbitt and roller bearings, and it was only after careful study of the temperature question with regard to all three types that satisfactory results were finally obtained.

Description of Testing Apparatus. The apparatus consists of 25 feet 10 inches of lineshafting in five equal sections, mounted in hangers which are inverted and used as floor stands. The hangers are bolted to two 8-inch I-beams which are leveled upon the floor. The shafts are of cold-rolled steel, $2\frac{7}{16}$ inches in diameter. Each section is 5 feet 2 inches long; the adjacent sections are coupled together by means of a flexible leather disk or two straps

connecting the two flange couplings. The flexible couplings prevent transmitting any part of the load applied on one shaft, to either adjoining section, and also prevent binding between shafts and bearings due to possible lack of alignment.

A direct-current Fort Wayne motor is directly connected to one end of the shafting by means of a flexible coupling. The motor is of the interpole type with the interpoles removed, making it a shunt motor. Its rating with the interpoles is $7\frac{1}{2}$ horsepower, 28 amperes, 400/1600 revolution per minute, four pole, 230 volts. The power required to run the motor alone at all speeds, without load, was accurately ascertained, as well as the power required to run the motor and shafts together, at all loads and speeds. The relative amounts of power required to overcome the friction of the various types of bearings were therefore accurately determined.

The load was applied through levers having hardened knife edges and pin points as fulcrums. Across the top of the 8-inch I-beams and at right angles to them, are bolted short 6-inch I-beams to which the fulcrums are attached. Standard 1000-pound scales are set upon the 6-inch I-beams. A double system of leverage is used in order to get sufficient load upon the bearings with as short a length of lever as possible. This double system of levers also serves to steady the apparatus and prevent excessive vibration. A pressure ratio of 8.33 at each bearing to one at the scale was obtained. This was checked by an independent method of weighing the actual load resulting at the bearings, from a given load on the scales. The loads were applied to the shaft by two bearings between each pair of hangers. These bearings are identical with those in the hangers, and are supplied with knife edges which engage a V-shaped groove in the 5-inch I-beam levers. The bearings and hangers for each section are symmetrically placed with respect to the middle of the section; therefore, equal loads on the intermediate bearings produce equal pressures on the end bearings. The reason for using twenty bearings was that the amount of power necessary for a single bearing was so

small as to be difficult of measurement; moreover any single bearing might not truly represent results that would be obtained from that type of bearing in general.

The Bearings Tested. The three kinds of bearings tested were: the Hess-Bright ball bearing manufactured by the Hess-Bright Mfg. Co.; the ring-oiled bearing manufactured by the Dodge Mfg. Co., lined with babbitt metal made from their formula; and the Hyatt roller bearing manufactured by the Hyatt Roller Bearing Co. All bearings were for the same size shaft and the same pieces of shafting were used for all the tests, except that two sections bent during the tests were replaced. The babbitt bearings are $9 \frac{21}{32}$ inches long and hence their projected area is 22.36 square inches. These bearings were oiled by the well-known ring-oiler device, there being two rings in each bearing. Each roller bearing contains six right-hand and six left-hand rollers, 0.780 inch in diameter; six are $9 \frac{9}{16}$ inches long and six are $9 \frac{3}{16}$ inches long. The bearings are of the type in which a cage is used for holding one-half the rollers. Each ball bearing contains a single set of balls $\frac{9}{16}$ inch in diameter. The diameter of the inner race across the ball groove is 3.4729 inches.

Mercury thermometers (two in each babbitt and roller bearing, and one in each ball bearing) were used for measuring the temperature of the oil or bearing. In order to avoid the endwise thrust of the shaft, when supported by the roller bearings, it was necessary to interpose two ball thrust collars. Before this was done, excessive vibration of the motor and of the apparatus resulted from the tendency of the shaft to move endwise. This was particularly troublesome at high loads and speeds.

Procedure when Making Tests. The power was measured by the ammeter voltmeter method. A second ammeter was used as a check on the first; and a watt-meter (arranged for direct and reverse readings) as a further check. The general order of taking data was as follows: Clean the commutator; adjust motor to speed; take all power readings; read all thermometers; adjust speed again and

repeat power measurements. This gave the power both before and after the temperature was taken, the mean of which would give the mean power for the mean temperature very closely. In addition, the readings acted as a check on each other.

The manner of making a test or run was essentially as follows: Each night the plant was run from three to twelve hours under the load and speed to be used during the run of the following day, but without observation. The purpose of the preliminary night run was to allow the shaft and bearings to adjust themselves to the conditions of the run. Then on the following day the shaft was run from three to six hours and frequent observations of power and

Relative Amounts of Power Consumed in Friction

Bearings	100 Feet per Minute		300 Feet per Minute	
	77 Deg. F.	100 Deg. F.	77 Deg. F.	100 Deg. F.
Ball	1	1	1	1
Roller.....	2.2	2.5	2.7	3
Babbitt	3	3.6	4.5	4

temperature were made during the run. The first few observations were made as often as practicable (about five minutes apart); the others, generally at fifteen-minute intervals, but toward the end of the run when the temperature was rising slowly observations were made at longer intervals which were generally of thirty minutes or more in duration.

The speeds used in the tests were between 150 and 450 revolutions per minute, corresponding, respectively, to about 100 and 300 feet per minute peripheral speed. Most of the loads used were between 700 and 1800 pounds per bearing, corresponding, respectively, to about 30 and 80 pounds per square inch for the babbitt bearings. All statements of results therefore are subject to the above limitations as to speed and loads. Two lubricants were used in all the tests: Atlantic red engine oil in the babbitt and roller bearings, and No. 2 Keystone grease in the ball bearings.

Relative Power Consumption. A comparison was made of the power consumed by friction in the babbitt, roller and ball bearings for bearing temperatures of 100 degrees and 77 degrees F., respectively. The power required for the babbitt bearings is higher than for the other bearings, except perhaps at low loads and speed, and the power for roller bearings is higher than for ball bearings. The excess of power for babbitt over rollers, and rollers over balls, increases with increase of speed for all loads. The table, "Relative Amounts of Power Consumed in Friction," shows the power consumed in friction by the three kinds of bearings at the speeds and temperatures indicated; the relative numbers are based, in each case, on the average power for

Comparison of Coefficients of Friction

Type of Bearing	Coefficient of Friction					
	Load, 727 Pounds		Load, 1227 Pounds		Load, 1727 Pounds	
	77 Deg.	100 Deg.	77 Deg.	100 Deg.	77 Deg.	100 Deg.
Ball	0.0025	0.0019	0.0022	0.0018	0.0020	0.0016
Roller	0.0069	0.0055	0.0055	0.0047	0.0049	0.0042
Babbitt	0.0112	0.0075	0.0082	0.0058	0.0070	0.0051

the three loads: 710, 1210, and 1710 pounds for balls; 740, 1240, and 1740 for rollers; and 730, 1230, and 1730 for babbitt.

The coefficients of friction for the three types of bearings, when subjected to average loads of 727, 1227 and 1727 pounds, respectively, and temperatures of 77 and 100 degrees F., are given in table, "Comparison of Coefficients of Friction." The peripheral speed was 150 feet per minute.

Lubricant Breakdown Tests. In order to observe the performance of the bearings under extraordinarily heavy loads, "breakdown tests" were run on each type of bearing with only one section of shafting on which were four bearings. This small number of bearings was used because it was impracticable to keep close watch of a larger number and avoid trouble during the excessively severe conditions.

The maximum load was 600 pounds on the scales, or about 5000 pounds per bearing. A speed of 200 revolutions per minute was chosen because it represents about the average lineshaft speed in practice. These tests began at about 3200 pounds per bearing. Failure occurred at about 4250 pounds per bearing in the case of the babbitt, 4650 pounds in the case of the ball bearings, and about 5100 pounds in the case of the roller bearings.

The quality and amount of lubricant used undoubtedly have an important effect upon the load that will cause a given bearing to fail. The bearings did not in any case fail structurally, as the power was cut off soon after distress was manifested, but the failure was simply that of the lubricant. Breaking down of the lubricant resulted in an immediate increase of the power required to maintain the original speed of rotation of the shaft in the bearings. In each case probably only one of the four bearings used in the breakdown tests showed distress at any one time. In the case of ball bearings, distress was manifested by disintegration of the grease which "melted" and ran out of the bearing. This was accompanied by the immediate increase in power requirement. Similar behavior on the part of the babbitt and the roller bearings indicated that at least one of the four under test was suffering from an approach to "metal-to-metal" contact. The bearings were not injured by these endurance tests, and all were used in subsequent tests at the more usual speeds and pressures.

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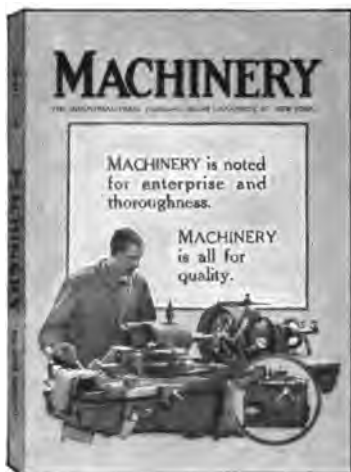
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